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NRX-A BLOCK I
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I. INTRODUCTION

After careful consideration of the various KIWI reactor geometries developed by Los Alamos Scientific Laboratory (LASL) the Westinghouse Astronuclear Laboratory (WANL) selected a reactor design based on the KIWI B-4A design for development as a flight type reactor. A major task of the NERVA program, then, is to adapt this KIWI design so that the NERVA reactor can meet the environmental, safety, and reliability requirements imposed during the shipping, handling, ground test, launching and flight phases of the program. The modifications made to the basic KIWI design are shown in the configuration which was presented in conjunction with the NRX-A design release of November 15, 1962. That release presented a core design which was basically the same as the KIWI B-4A with the addition of a stiffer core lateral support system and a core nozzle end seal. The other changes included longer tie rods, different tie rod material, multiple tie rod clusters at the core periphery, thicker support blocks, a revised system of supporting the core assembly and outer reflector and addition of Class A locking devices. A lateral support system was incorporated to restrain the core under steady and dynamic loading conditions so that the reactor core could withstand environmental shock and vibration loads which were anticipated at that time. In addition, the forward end seal design that was featured in the KIWI configuration was changed to an aft-end location and the core seal was redesigned.

Late in November 1962, the KIWI-B4A suffered severe mechanical damage during test. Discussions between WANL and LASL personnel concerning the apparent problems, coupled with the results of analyses and investigations conducted at WANL, resulted in a number of revised design objectives for the NRX-A core periphery. The major objectives were as follows:

(1) Limit access of cool gas to the core periphery to prevent leakage into the core, thereby reducing potential thermal stress problems.

(2) Make certain that the pressure between fueled elements is no higher than the external pressure on the bundled core, thereby eliminating element separation and fluttering.

(3) Permit axial and radial thermal expansion of the core and peripheral components.

(4) Develop a design sufficiently flexible to accommodate changes in the bundling load applied to the core without concomitant major changes in components.

The NRX-A reactor incorporating the changes which came about as a result of the KIWI-B4 test and related investigations was presented in a March 1963 design release. This design, as modified by some major changes that have been defined since that period, constitutes the Block I configuration which is discussed in this report.

Essentially, the report discusses the reactor "from the outside in," i.e., it begins in Chapter III with a discussion of the interface areas and proceeds "inward" to the analysis of the shield (Chapter IV), the Flow Screen (Chapter V), the Outer Reflector (Chapter VI), and the Core Assembly, which also includes detailed discussion of the Inner Reflector (Chapter VII).

For ease of reading, the information given in each chapter is presented under five headings: Design Philosophy, Description, Material Selection, Design Analysis, and Environment.

In order to round out his understanding of the design, the reader should review the Mechanical Test and Data Requirements, WANL-TMI-581, which details the data requirements for the NRX-A1 cold flow test, and the Shipping Demonstration Shock and Vibration Analysis, Volumes I and II, WANL-TME-458. This latter report evaluates the design adequacy of the NRX-A shipping container. In addition, the detailed analytical information which a reader might want to know about a given component can be found in one of the numerous analyses reports referenced in the Bibliography.

II. BASIC CONFIGURATION AND OPERATIONAL CRITERIA

A. BASIC CONFIGURATION

This chapter considers four basic features of the NERVA Block I reactor: reactor make-up and function of each part, development of realistic environmental criteria, development of realistic stress criteria, and the functional sequence of reactor operation.

The NERVA Block I reactor is a nuclear engine that can fit inside a pressure vessel which has an inside diameter of 49.75 inches and is 91.973 inches long (measured from the nozzle flange face to the inside of the dome). A half-scale mockup of this reactor is shown in Figure 2-1.

The reactor is composed of a shield, which reduces propellant heating and restricts heating of the engine components located between the pressure vessel and the propellant tank; the flow screen, which filters the coolant flow to make certain that the fuel orifices are not clogged; the outer reflector, which reflects the neutrons back into the core (in order to conserve neutrons and permit criticality with a smaller total mass) and houses the control drums which control the reactivity of the assembly; and the core assembly, which is composed of the core proper and a neutron reflecting core-lateral support area that is commonly called the inner reflector. The core proper serves two basic function: (1) it contains the fuel, provides the heat source, and acts as a heat exchanger to add heat to the rocket propellant; (2) it acts as a nuetron moderator. The inner reflector structure also performs four incidental or subsidiary functions: (1) it absorbs and transmits axial and lateral loads; (2) it seals the core against bypass leakage; (3) it aids in maintaining the proper bundling pressure; and (4) it distributes coolant flow for proper heat dissipation.

B. OPERATIONAL CRITERIA

The maximum dynamic environmental conditions which the reactor must meet for each of the specified operational modes are clearly defined in Table 2-1, Summary of NERVA Reactor Environments. These "maximum permissible limits" are compared against anticipated reactor vibration levels for ground handling (Figure 2-2) and boost (Figures 2-3a and 2-3b). Figure 2-4 shows the anticipated sound level during the boost phase.

In defining the NERVA design, attention was also given to the development of the stress criteria which were needed to measure the various stresses which the reactor could successfully stand. These criteria are given in detail in WANL-TME-512, NRX-A Stress Criteria. Three of the more salient areas have been extracted for this report because they present the "overview" considerations which are closely related to this discussion of basic reactor configuration.

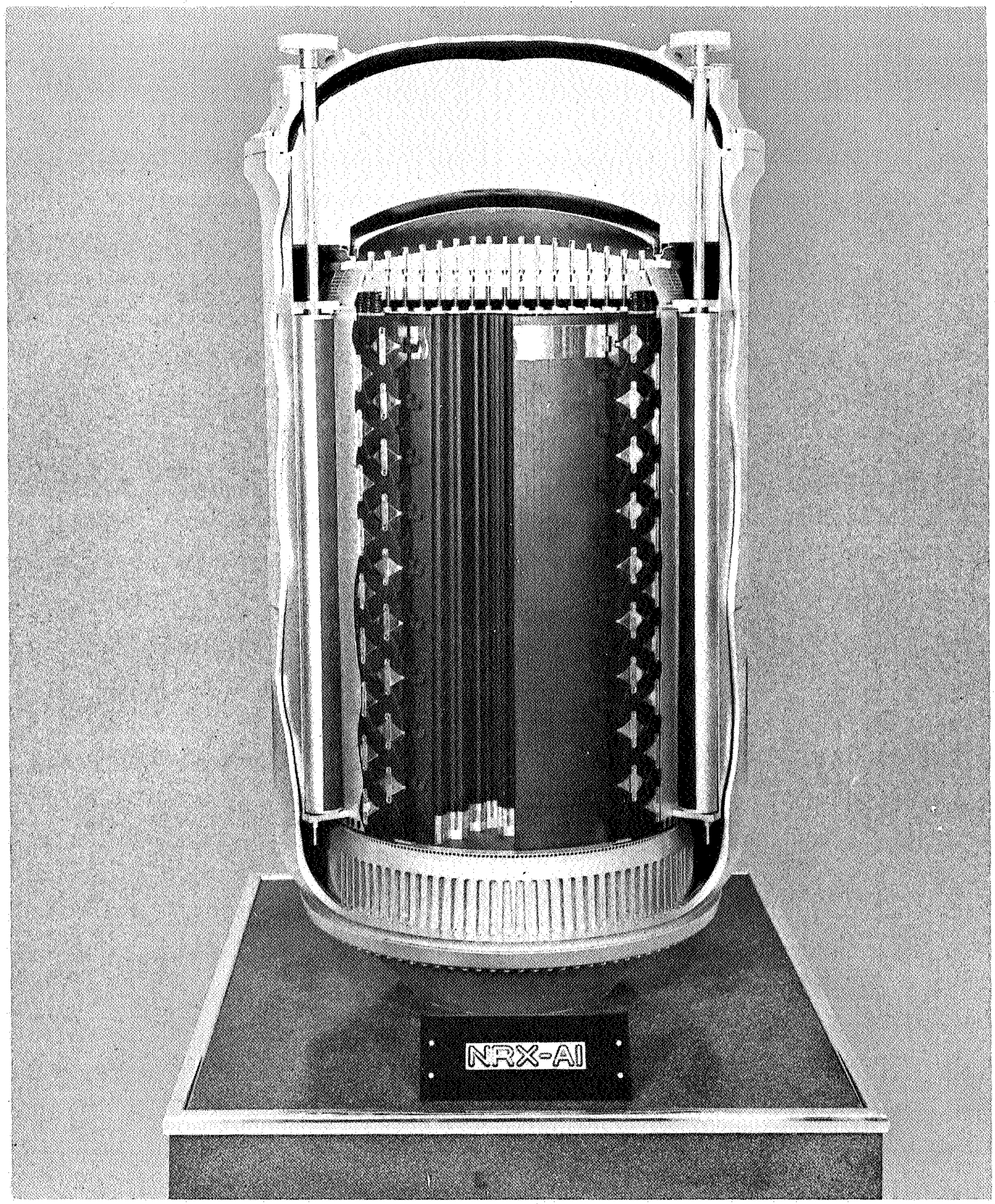

FIGURE 2-1. NRX-A MOCKUP: HALF SCALE

TABLE 2-1

Summary of NERVA Reactor Environment

	Shipping	Ground Handling	Static Test	Booster Operation	Nerva Engine Operation
1.0 Sustained Loading					+2.0 g, -1.2 g parallel to thrust axis.
1.1 Linear Accelerations	±1.0 g parallel to thrust axis, zero normal to thrust axis	1.5 g along any axis.	±1 g parallel to thrust axis, zero normal to thrust axis.	+5.0 g, -1.2 g parallel to thrust axis, ±1.0 g normal to thrust axis.	±1.0 g normal to thrust axis.
1.2 Spin Acceleration and Velocity	Negligible	0.010 rad/sec^2 and 3 rpm	Not Applicable	Negligible	Negligible
1.3 Angular Acceleration and Velocity in Pitch and Yaw Modes	Negligible	Negligible	Negligible	15 deg/sec^2 and 10 deg/sec	15 deg/sec^2 and 10 deg/sec
2.0 Dynamic Loading		+2.0 g, -1.2 g parallel to thrust axis.			
2.1 Shock	±2.5 g axially, ±1.8 g laterally, long duration.	±1.0 g normal to thrust axis, maximum duration 0.01 sec.	.5 g along any axis, maximum duration 0.010 sec.	1 g along any axis, maximum duration 0.010 sec.	1 g along any axis, maximum duration 0.010 sec.
2.2 Vibration	±1.7 g axially, ±1.0 g laterally, between 2.5 and 7.5 cps.	Figure 2-2	Figures 2-3 SPL 161 db	Figures 2-3a and 2-3b	Figures 2-3 & 2-4
3.0 Ambient Conditions		650 to 795 mm Hg	650 to 795 mm Hg		
3.1 Pressure	5 PSIG	Rain, sand, and dust	Rain, sand, and dust	30-inch Hg to -10^{-8} mm Hg.	10^{-8} - 10^{-12} mm Hg
3.2 Temperature	-80 °F to +160 °F	-80 °F to +160 °F	-80 °F to +160 °F	-80 °F to +160 °F	External Heat Supplied By Radiation Only
3.3 Humidity	0 to 20%	0 to 100% + salt atmosphere	0 to 100%	0 to 100%	Zero

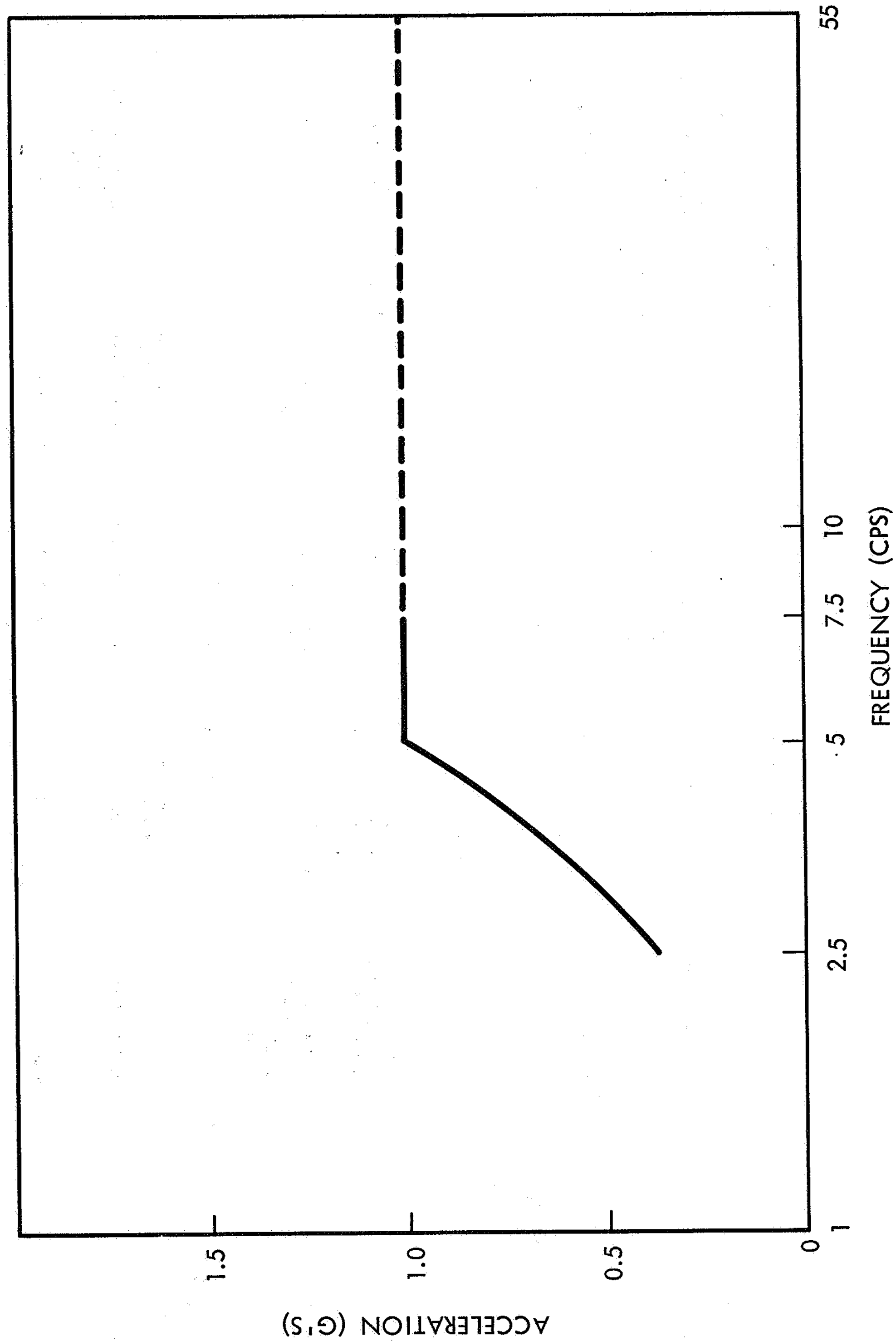

FIGURE 2-2. ANTICIPATED VIBRATION DURING GROUND HANDLING

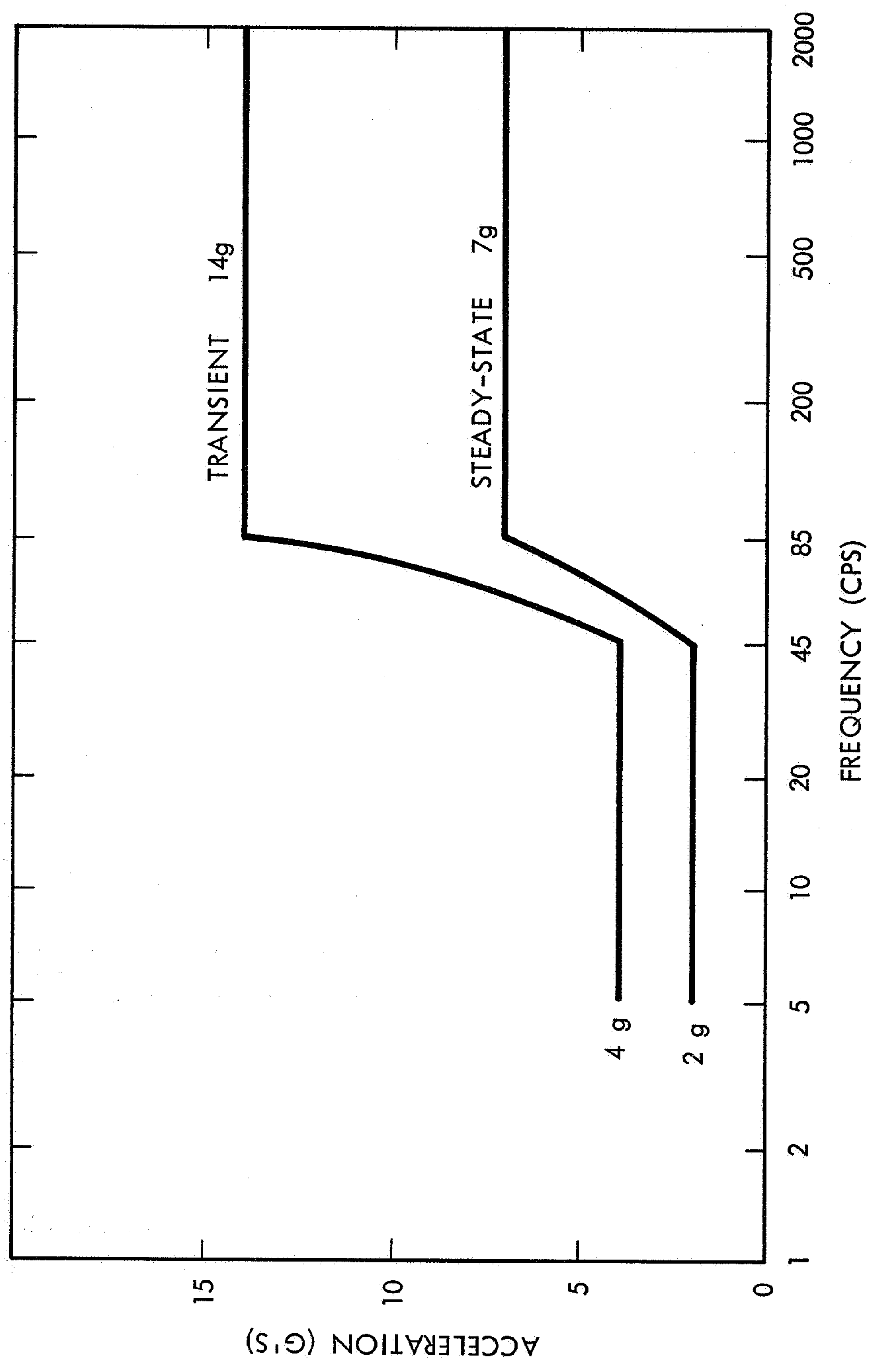

FIGURE 2-3a. ANTICIPATED VIBRATION DURING BOOST

NOTE: DATA FROM MARSHAL SPACE FLIGHT CENTER

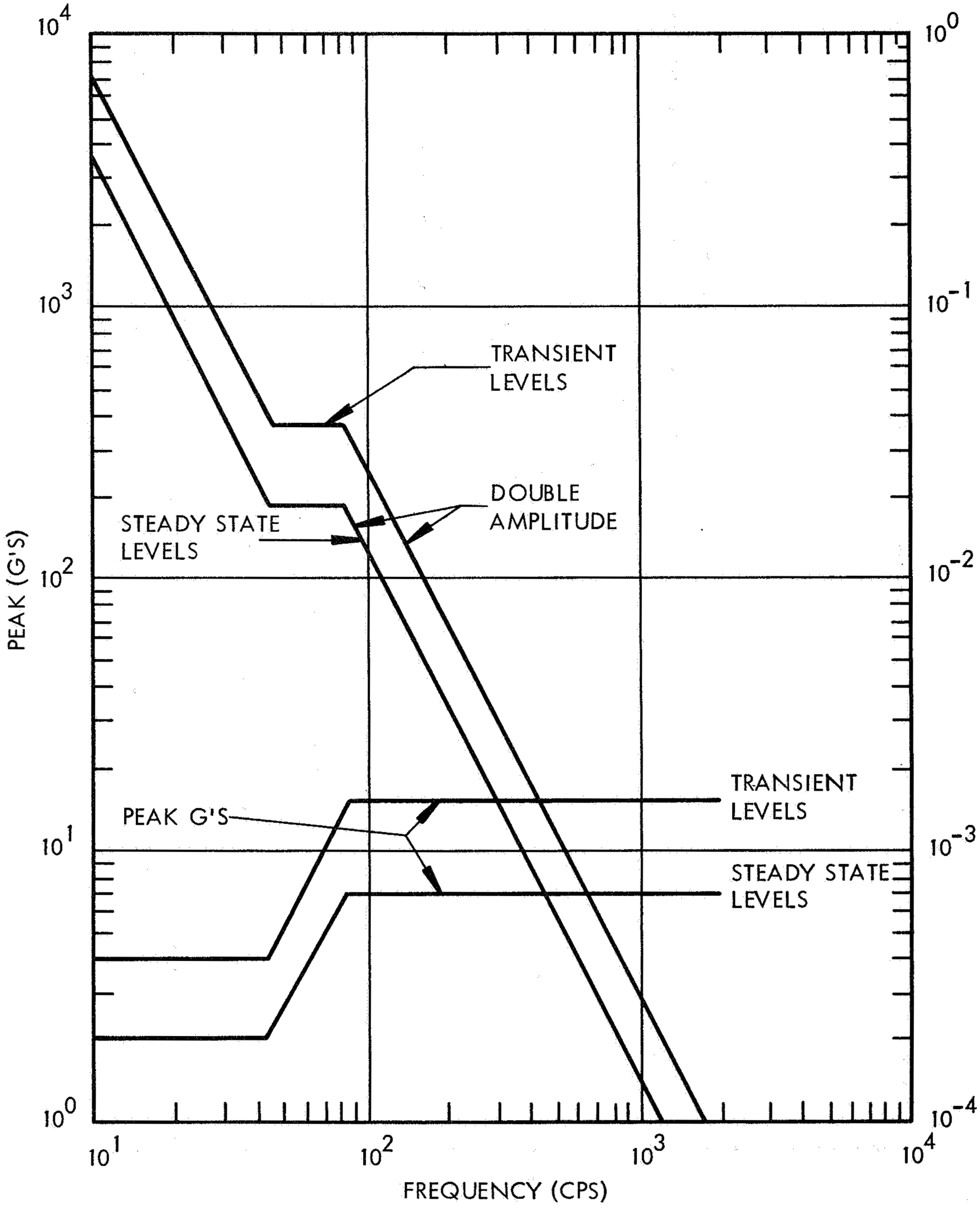

FIGURE 2-3b. ANTICIPATED VIBRATION DURING BOOST

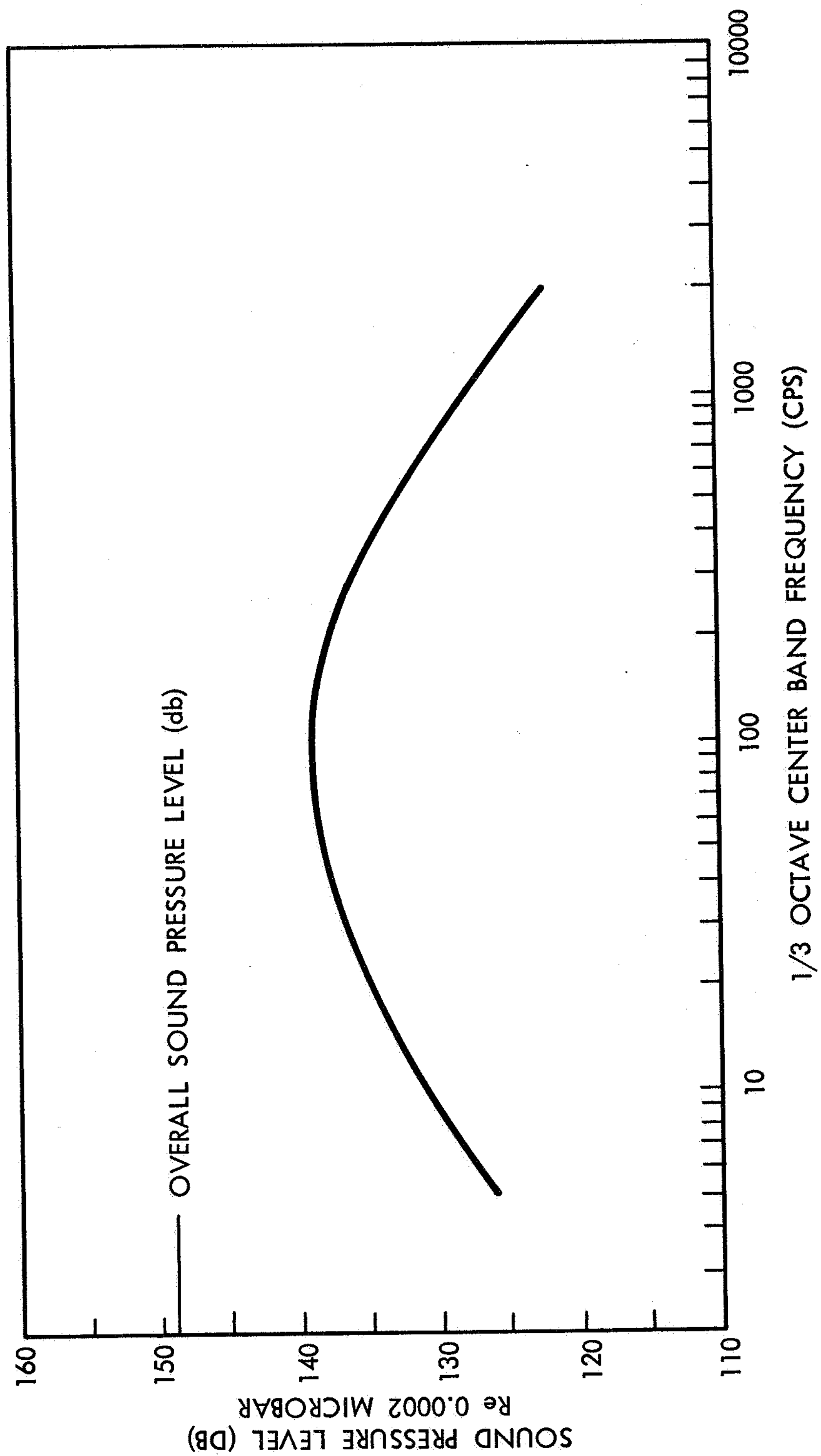

FIGURE 2-4. ANTICIPATED SOUND PRESSURE LEVEL DURING BOOST

1. Design Pressure Drop

The design pressure drop is the normal calculated difference in the coolant pressure across two points in the reactor, multiplied by a load factor. The load factor should be based on a reasonable conservative deviation from the normal calculated value or combination of values which will result in a more serious loading condition for the component considered. Ordinarily, such design pressure drops will be within 5 to 10% of the calculated normal pressure drops. However, the pressure drop will sometimes vary, depending on the uncertainty of the assumptions included in the calculations and the amount of monitoring to be employed during testing.

2. Design Temperature and Temperature Gradients

These items are defined as the normal calculated temperatures and temperature gradients in the reactor components, adjusted by some value. This adjusting value shall be based on a reasonably conservative deviation from the calculated temperature or temperature gradients or a combination of temperature or temperature gradients which will result in a more serious design condition for the component considered, such as higher deflection, higher stress or lower strength. While such temperature deviations will ordinarily be within 50° R of the calculated temperatures, the figure will depend in large measure, on the degree of uncertainty in the calculations.

3. Stress Design Criteria

The maximum shear or distortion energy theories shall be used as a basis in designing ductile materials. When referring to the strength of a given material (yield, ultimate, fatigue, etc.), the minimum strength at design metal temperature based upon minimum specification values shall be used. The allowables to which the steady-state component stress must be composed are listed below.

Membrane Stress. The stress based upon the average value of primary stress across the minimum thickness of a section, neglecting structural discontinuities and stress concentrations caused by design values of internal pressure, mechanical forces, or their combinations. They shall not exceed 80% of yield strength or 60% of ultimate strength at design temperatures, whichever is less. This value is equivalent to a 1.25 safety factor based on yield, or 1.67 on ultimate.

Peak Stress. The primary plus the secondary stresses which are exerted at any location due to design pressure, mechanical forces, pipe reactions, or their combinations, including the effect of structural discontinuities, but excluding stress concentrations. At design temperatures, these stresses shall not exceed 100% of yield strength or 80% of ultimate strength, whichever is less.

Thermal Stress. Thermal stress is the stress caused by external constraint, incompatibility of natural expansions or, contractions of different areas of the body which are caused by temperature conditions. The analysis of thermal stresses in a structure will frequently result in the calculation of misleading stress values which are beyond the ultimate strength of the material. These values are misleading in that the design is still satisfactory since the analysis does not account for stress relaxation at various temperatures. To account for this, all thermal stresses shall be considered as transient conditions and treated accordingly. In addition, steady-state-thermal stresses shall be limited by the following requirements:

(1) Distortion limitations on dimensions and clearances which are necessitated by safety and operational conditions shall not be exceeded.

(2) When, at operating temperatures, the peak stress acting in combination with the steady-state thermal stress (including the effect of structural discontinuities, but not stress concentration) results in a stress which exceeds the limit of elastic behavior, justification for this stress level must be provided. This justification must show that no deleterious effects, such as those due to distortion, creep, or stress rupture, will occur in either the new or irradiated condition of the material.

C. FUNCTIONAL SEQUENCING

In order to view the parts of the NRX-A reactor as they are functionally sequenced, the flow path will be traced as it progresses through the reactor (Figure 2-6). Because this report is primarily concerned with the basic parts of the reactor, most of the interface areas are avoided and the flow path is picked up as it emerges from the nozzle. Furthermore, no attempt is made to describe the control or feed-back mechanism as this interface area is physically outside the reactor. Figure 2-5 calls out all of those areas through which the coolant is channeled.

1. Nozzle-Pressure Vessel Plenum

As the coolant exits from the nozzle into the nozzle pressure vessel plenum it divides into four flow paths: the outer reflector-pressure vessel annulus, the outer reflector, the inner reflector-outer reflector annulus and the inner reflector. Coolant flow is

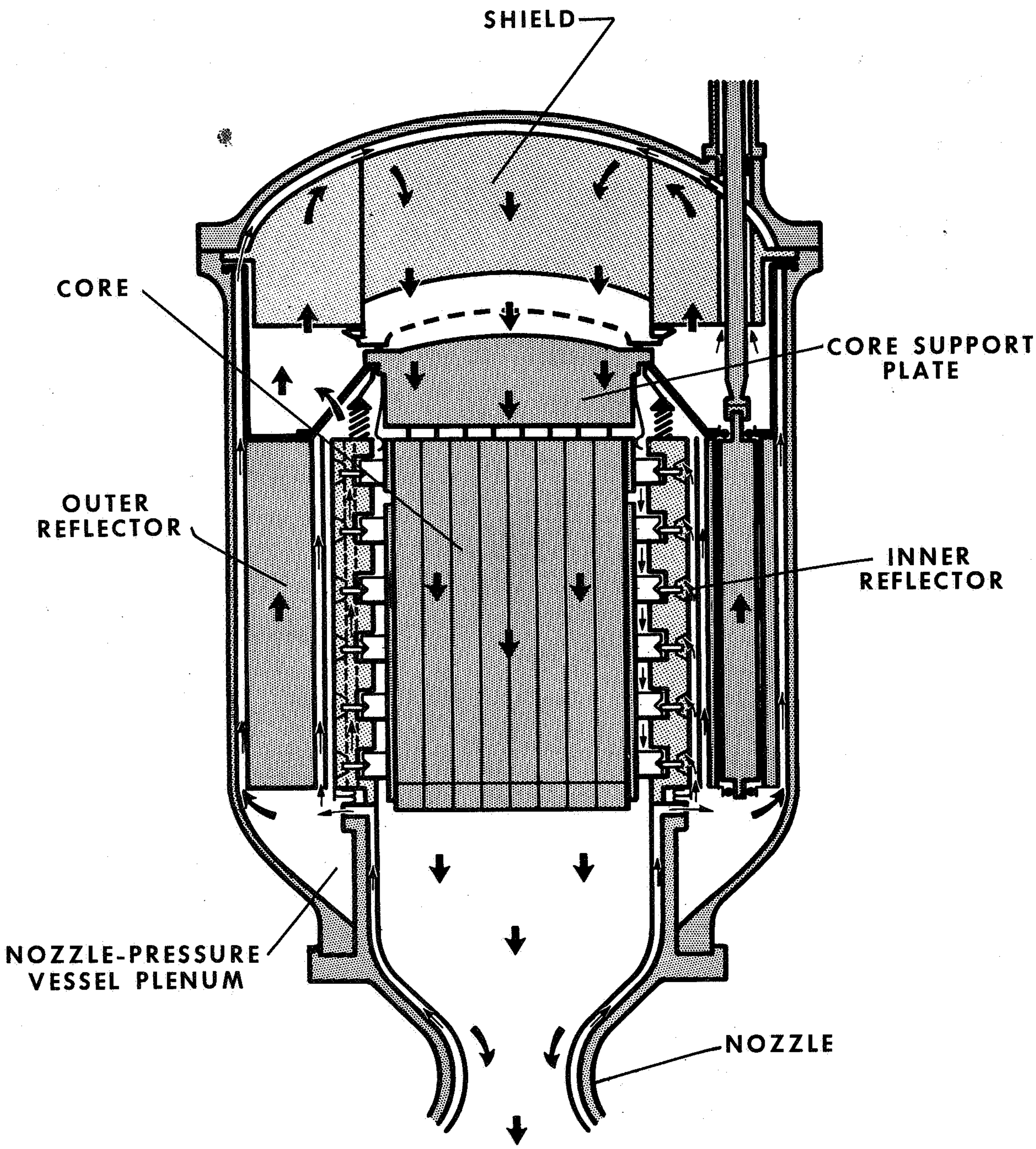

FIGURE 2-5. TWO PASS FLOW SYSTEM

prevented from escaping into the nozzle exhaust plenum by a seal which bridges the interface between the nozzle and the graphite inner reflector cylinder. Holes are drilled through the graphite cylinder to assure that any leakage through this seal exists to the exhaust plenum. This arrangement ensures a predetermined pressure drop of approximately 190 psi which is sufficient to force the self-energizing seal to seat.

2. Outer Reflector

The coolant enters the outer reflector-pressure vessel annulus by passing through the annulus between the outer diameter of the nozzle end outer reflector support ring and the pressure vessel. This annulus is interrupted in 24 places by the projections on the nozzle end support ring which center the aft end of the outer reflector in the pressure vessel. In addition to centering the reflector, these projections also guarantee a minimum flow area at the inlet to the annulus. After passing through this annulus for the entire length of the reflector, the coolant flows through 360 metering holes in the support flange of the dome end support ring and exhausts into the plenum between the shield and the pressure vessel dome.

Coolant enters the outer reflector through holes in the nozzle end support ring. After passing through these holes, it enters a 0.125-inch plenum and is redistributed to the 192 coolant holes in the sectors. When it leaves the sectors, it again enters a 0.063-inch plenum and is redistributed to holes in the dome end support rings. It then enters the plenum between the aft side of the shield and forward end of the outer reflector, where it mixes with the coolant passing through the annulus between the inner and outer reflectors.

The control drum flow system parallels the sector system in the following manner. Coolant enters the control drum by passing through holes in the bearing housing (which is integral with the nozzle end support ring) through the nozzle and bearing and through the nozzle and bearing shaft. The coolant then flows through the beryllium cylinder, over the control plate and into the annulus between the control drum and the sector. There are four independently-orificed channels; the outer row of holes in the beryllium cylinder, the inner rows of holes in the beryllium cylinder, the control plate annulus, and the annulus between the control drum and the sector. All orificing is accomplished in the dome and bearing shaft. After passing through the control drum, the coolant exits through dome and bearing housing and bearing and enters the shield-outer reflector plenum. At this point it mixes with the coolant passing through the sectors and the coolant passing through the inner reflector.

Coolant enters the annulus between the outer reflector and the inner reflector through the annulus between the inside diameter of the nozzle end support ring and the outside diameter of the aluminum barrel. Spacer strips on the OD of the aluminum barrel center the core assembly in the outer reflector and assure a minimum annulus for coolant flow. The coolant flowing through this annulus is orificed by a metering ring which is located near the forward end of the aluminum barrel. After passing through the core support ring, this coolant also enters the shield-outer reflector plenum.

At this point, the coolant passes through the shield in a forward direction, completing the first pass of a two pass system.

3. Inner Reflector

The coolant passes through axial holes in the graphite cylinder and through channels routed into the outer surface. These channels provide a path for the coolant from the aft to the forward end of the cylinder. Channels are provided in the cylinder surface to route the coolant between the aluminum barrel and cylinder to the laterial support springs in order to maintain proper temperature in the springs. One of the primary purposes of the graphite cylinder is to house the laterial support system. The lateral support system, in turn, serves three purposes: First, it supports the reactor core and absorbs shocks and vibrations; second, it provides core periphery sealing which controls coolant pressure distribution and; third, it provides adequate bundling pressure on the core. In order to accomplish this sealing, 18 segmented ring seals, housed in grooves on the inside of the graphite cylinder, surround the core. Each of the rings is composed of 12 graphite segments. Two of the rings serve to retain the filler strips and core band, while the other 16 actually serve as seals. The graphite segments are held against the core periphery by the action of graphite plungers working through the cylinder under the force of the lateral support springs.

The support ring assembly, which is located on the forward end of the graphite cylinder, serves as a support and locating device for the preload springs. These springs transmit axial loads from the cylinder into the flange of the core support ring. Holes and slots in the core support ring allow the coolant flow to empty into the outer reflector-shield plenum and combine there with the coolant coming from the outer reflector as it enters the first pass of the shield.

4. Shield

Once the coolant completes the first pass through the shield it empties into the pressure vessel dome-shield plenum. The second pass is made through the flow holes into the shield-core support plant plenum. This shield, coupled with the core support plate-to-shield seal, makes the two pass system possible. It should be noted that the flight version of this unit will be a lithium hydride, stainless steel unit which has the neutronic shielding capabilities required for flight operation.

5. Core Support Plate

From the shield-core support plate plenum, the flow passes through the core support plate which is made of 2219 forged aluminum. The plate has approximately 2500 holes drilled axially through it. These holes serve two purposes: they allow the coolant to pass into the core support plate-shield plenum, and they supply a means of inserting and attaching the fuel cluster tie rods. Both negative and positive axial loads are reacted through the core support plate. These loads are transmitted through the flange of the support ring assembly to the outer reflector dome and support ring.

6. Core

The final portion of the flow is through the reactor core. At this point, the coolant gas absorbs the large amount of heat needed to produce significant thrust. This gaseous coolant passes through the holes in the fuel elements, and a carefully-metered fraction is allowed to pass through the non-fueled elements from where it exits into the exhaust nozzle. The reactor core is composed of fueled and unfueled elements, tie rods, support blocks, filler strips, and associated hardware.

The fuel elements are hexagonal shaped graphite rods. A graphite material is extruded with the fuel (pyrolytic graphite coated uranium dicarbide beads) evenly dispersed throughout the matrix. The elements are extruded with 19 coolant holes and are grouped to form both the regular and irregular clusters. A regular cluster has a center unfueled element surrounded by six fueled elements. At the forward end, these elements are held in a cluster by the cluster plate and related hardware. The support block on the aft end of the cluster acts as an axial support for the clusters. The entire assembly is hung from the core support plate by the tie rod and its associated hardware.

Because these clusters do not form a smooth circular periphery when they are grouped together, irregular fuel clusters must be added. These irregular clusters are combinations of fueled, unfueled and partial fueled elements. These elements are grouped in a series of patterns (seven distinct configurations) which round out the core periphery. Filler strips complete the transformation to a smooth circle. These strips are also used to retain the insulating tiles by sandwiching them against the core and providing a ledge by which the tiles are axially retained. These pyrolytic tiles are used to prevent excessive thermal gradients and thermal stresses from being exerted on the fuel elements and filler strips.

III. INTERFACE AREAS

The design of the NRX-A reactor and reactor control system is the responsibility of Westinghouse Astronuclear Laboratory (WANL), while the nozzle, pressure vessel and other areas are being designed by Aerojet General Corporation (AGC). Because there is a direct relationship between the reactor and these other areas, there are mating surfaces or interfaces which must be defined and co-ordinated. These mechanical interface areas are at the instrumentation port, control drum drive shaft port, the dome end support ring flange, the nozzle interface seal, and the nozzle end support ring (Figure 3-1). These areas are discussed in this chapter both to outline interface problems and to indicate how these problems have been solved.

A. NOZZLE END SUPPORT RING

The nozzle end support ring is one of the two titanium rings that enclose the 12 beryllium sectors that comprise the outer reflector. This support ring contains the nozzle end control drum bearings and serves to position and restrain the radial motion of the nozzle end of the beryllium reflector. While the ring is designed to restrain the nozzle end of the beryllium reflector, at the same time, it also allows coolant to flow in the annulus between the outer reflector and the pressure vessel.

The actual interface area is that area where the support ring and the pressure vessel are in contact. The stops or losses (projections on the outer circumference of the ring) bear against a raised machined ring in the pressure vessel and restrain lateral movement of the nozzle end of the outer reflector. The scalloping between the stops or losses along the outer circumference of the ring provides a minimum passage for coolant flow, which is required to cool the pressure vessel. The coolant enters through these scallops into the outer reflector-to-pressure vessel annulus.

Further details concerning the nozzle end support ring can be found in Chapter VI which is devoted to the Outer Reflector Assembly.

B. NOZZLE INTERFACE SEAL

The interface seal between the nozzle and graphite inner reflector is composed of two flexible U-shaped rings which are made of Inconel X-750 metal, welded to a central flanged ring which is made of the same material. The seal provides an elastic face between the graphite cylinder and the nozzle surface and prevents the coolant flow from by-passing the core, by escaping from the nozzle tube bundle directly to the core exhaust plenum.

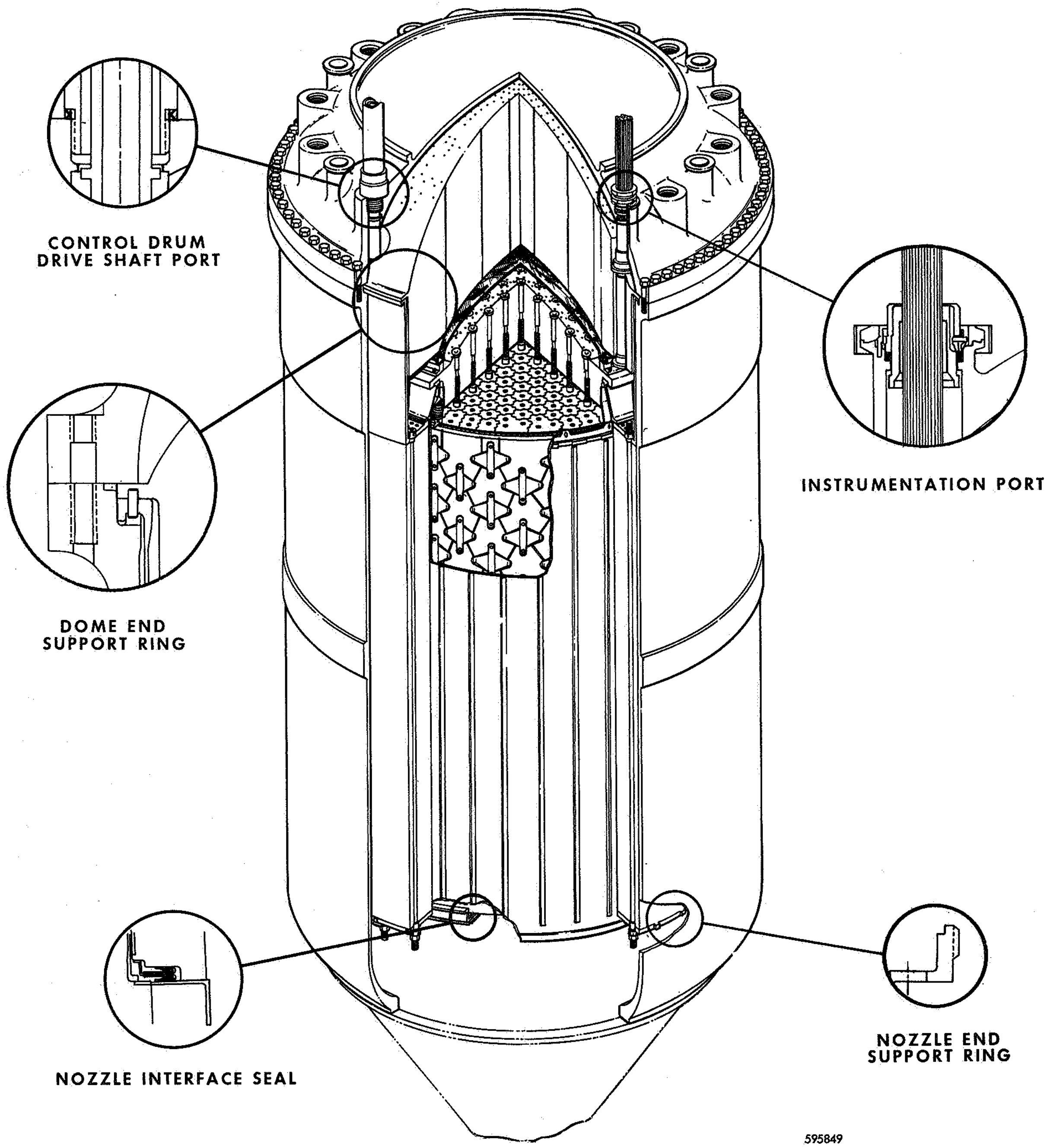

FIGURE 3-1. NRX-A INTERFACE AREAS

This seal flexibility prevents leakage that could occur at this plane because of mismatch between the nozzle and graphite barrel. When the seal is in place (Figure 3-1) with the nozzle attached, it is compressed within the confines of the groove at the nozzle end of the graphite cylinder. This seal is both pressure and spring activated. Leakage bleed-off ensures that the full pressure drop from the reactor inlet-to-core exhaust is always exerted on the seal. This pressure tends to spread the U-shaped sections, forcing them against the sealing surfaces.

Further details concerning the nozzle interface seal are presented in Chapter VII, which discusses the graphite cylinder and the nozzle end seal in detail.

C. DOME END SUPPORT RING

The dome end support ring transmits the entire reactor load to the pressure vessel at the dome end flange. It is composed of a z-shaped, titanium ring and is bolted to the end surfaces of the 12 beryllium sectors which comprise the outer reflector. The whole assembly is clamped between the dome end of the pressure vessel and the flange.

The physical interface area is located at the support ring flange (Figure 3-1). This interface is the point where the dome end support ring flange, the shield support flange, and the pressure vessel cone are in contact. The coolant holes in the flange allow the coolant to pass from the pressure vessel-outer reflector annulus into the pressure vessel dome-to-shield plenum, to join the main coolant stream as it begins the second pass through the shield.

A dowel pin, which is press fitted into the dome end support ring flange, extends into a groove in the pressure vessel flange and through the shield flange into the closure flange. This pin is used to orient the dome end support ring and align the coolant holes in the shield support flange.

Further details on the dome end support ring can be found in Chapter VI, which is devoted to the Outer Reflector Assembly.

D. CONTROL DRUM DRIVE SHAFT PORT

The fourth physical interface area is at the control drum drive shaft port. There are 12 of these ports, each providing a passage for the actuator shaft through the pressure vessel.

The actuator shaft has an extended shoulder which is positioned at the port opening on the inside of the pressure vessel in such a manner that only a minimum of coolant will pass around the shoulder and through the port. This interface area is designed to limit the amount of coolant that will escape from the pressure vessel dome-to-shield plenum should the plug develop a leak. The coolant that does pass through the port is confined to the guide tube which surrounds the section of the drive shaft extending from the drive shaft port to the privy.

The actual interface is the guide tube which is threaded into the closure. It contains a shoulder that confines a seal against the closure port. The opposite end of the guide tube penetrates the privy roof and passes through a floating "O" ring seal.

E. INSTRUMENTATION PORT

The fifth interface area is located at the instrumentation port on the pressure vessel dome. These ports are used to provide an exit for the reactor's instrumentation leads (Figure 3-1).

A "Bar-X" seal, which is located on the instrumentation support assembly and is covered with a metal spacer, provides sealing of the instrumentation port in the dome. A nut, screwed on the protruding end of the instrumentation support assembly, draws the entire unit together. This exerts a force on the seal which then expands radially and prevents coolant from escaping from the pressure vessel dome-to-shield plenum.

The instrumentation leads are brazed at the instrumentation support assembly outlet. This brazing forms a seal which prevents coolant leakage around the instrumentation leads.

IV. NRX-A SIMULATED SHIELD

A. DESIGN PHILOSOPHY

The KIWI B-5 shield represented the first generation in the development of a shield for the nuclear rocket engine. The shield is located internal to the pressure vessel between the reactor core and the propellant tank. The primary purpose of this shield is to reduce both the nuclear heating of the propellant and the activation and heating of the engine components located between the pressure vessel and the propellant tank. As a component of the nuclear rocket system, the shield is subject to the following environmental conditions:

(1) Flight and ground test loads consisting of sustained and dynamic accelerations, plus acoustically-induced vibration.

(2) Temperature and pressure-induced loads.

(3) Ground handling loads.

In light of these requirements, the initial design objective was to produce a shield which would be suitable for all of the environmental factors, even though the KIWI B-5 shield would not be subject to sustained flight accelerations.

The initial design effort was concentrated on this flight-type shield, and detailed drawings for the NRX-A counterpart were released in November 1962. As a result of the program redirection in the spring of 1963, the work on the flight-type shield was terminated and the shield design effort was then directed toward developing a substitute unit for the initial ground test reactors (Figure 4-1). All detail drawings for this substitute unit were released in August 1963. Since this is the shield configuration which will be used with the Block I reactors, the balance of this chapter will concentrate on an analyses of this shield.

B. DESCRIPTION

The NRX-A simulated shield is a domed structure. The forward end surface is a 2 to 1 convex ellipsoid, while the aft end is a matching concave surface.

The envelope of the cold flow shield is the same as the envelope of the hot flow shield, except for the plenum at the forward end of the core support plate. This plenum appears to be

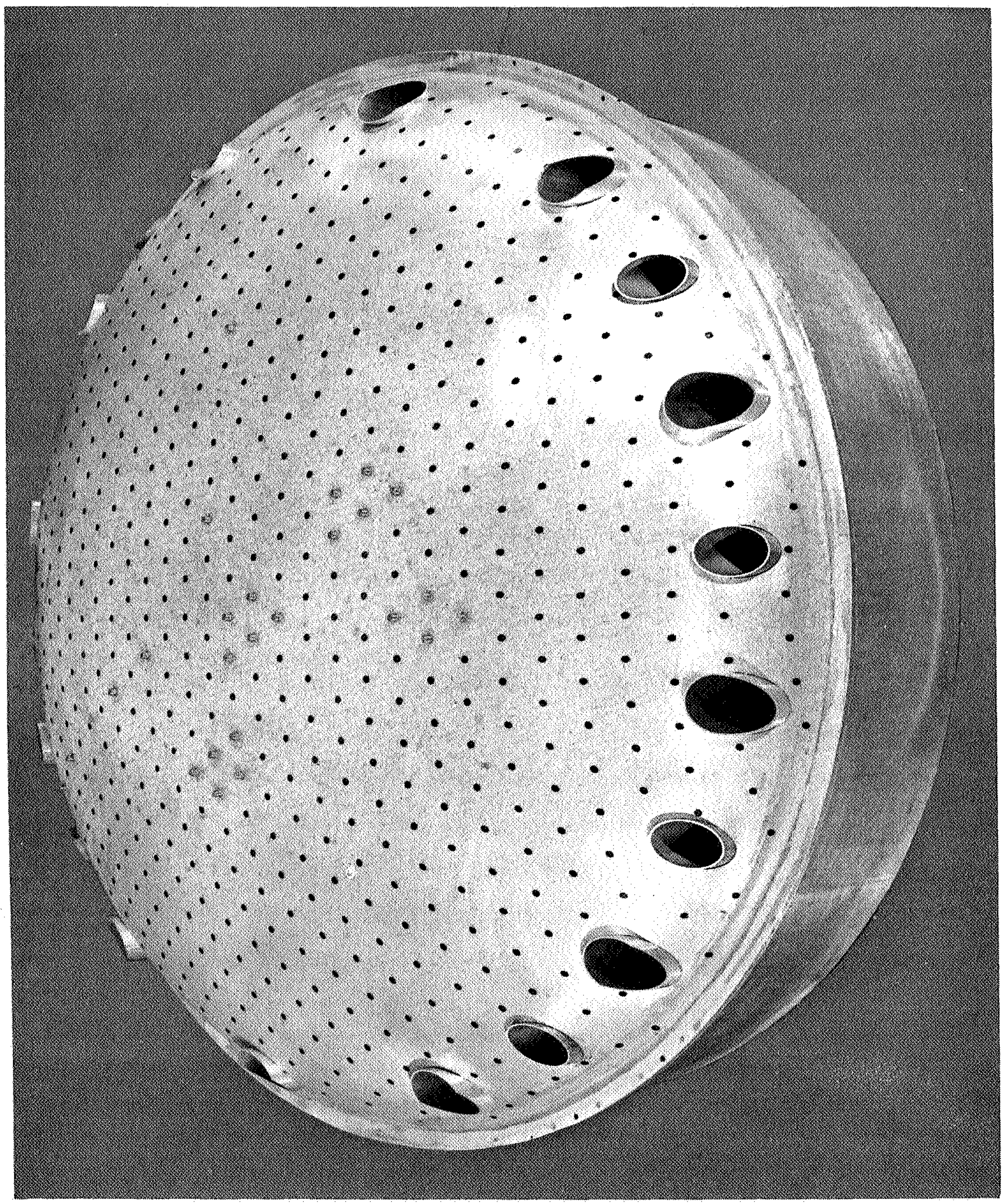

FIGURE 4-1. NRX-A SIMULATED SHIELD

larger than the plenum between the hot flow reactor shield and the core support plate; however, this is due to the fact that the reactor shield has an internal plenum, whereas the simulated shield does not. The total reactor shield plenum dimensions were matched in the simulated shield design to minimize any maldistribution of flow that might result from gross changes in dimensions in the inlet plenum to reactor core.

There are 589 coolant holes, 0.50 of an inch in diameter, extending axially through the shield. The flow path, Figure 4-2, is a two-pass system, the same as that in the hot reactor shield. In the reactor assembly, the coolant passes through the inner and outer reflector in a direction from the nozzle end toward the dome. It is directed through the first pass into a plenum between the shield and the pressure vessel dome. From this plenum, the coolant enters the second shield pass from which it exits into the core inlet plenum. A seal between the reactor core and the shield keeps the reactor inlet coolant from mixing with the shield inlet coolant.

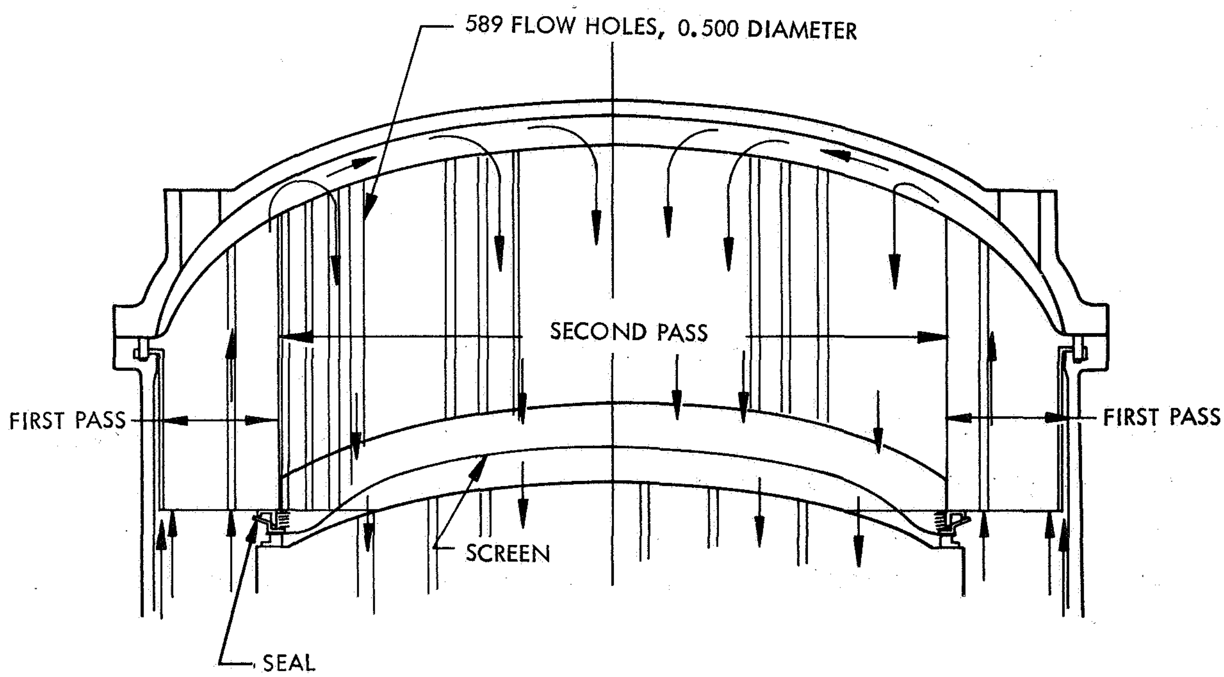

Figure 4-2. Flow Path Through Shield

Except for a modification of the attachment, the seal between the simulated shield and reactor core is identical to the one used for the hot reactor shield. The screw holes used to attach the seal are drilled oversize, and the seal is loosely attached. This arrangement prevents undue stresses which could be caused by a difference in thermal expansion between the simulated shield and the steel seal flange.

An integral flange around the periphery provides structural support for the shield; i.e., the outer diameter of the flange fits closely against the pressure vessel and resists lateral movements. A single dowel pin which is inserted through the flange and into the pressure vessel orients the shield angularly (Figure 4-3).

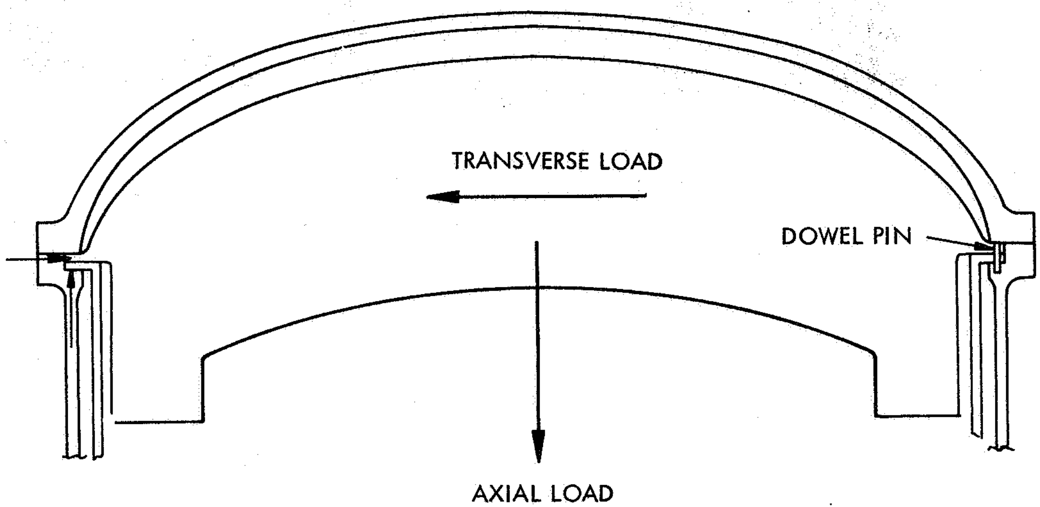

Figure 4-3. Shield Structural Support

In the dome-up configuration, three lifting brackets, bolted to the forward end surface of the simulated shield, are used for handling and installation. In the dome-down position, shield handling is aided by the use of tapped holes in the aft end surface of the aluminum shield. Eye bolts installed in these tapped holes perform the same functions as the lifting brackets.

C. MATERIAL SELECTION

The simulated shield used for the NRX-A Block I tests is machined from a solid 2219 aluminum forging. This shield, which will be used for both the NRX-A1 (non-nuclear, gas flow) and the NRX-A2 (nuclear or hot) test series, has the following functional requirements:

(1) It provides the same flow direction to the coolant as that provided in the reactor shield.

(2) It matches the flow impedance of the reactor shield.

The following critical pressure distributions will impose a load on both the simulated shield and the seal which is located between the reactor core and the shield:

Pressure at inlet of the shield	701.0 psia
Pressure loss across Pass 1	6.4 psi
Pressure at Pass 2 inlet	694.6 psia
Pressure loss across Pass 2	6.6 psi
Pressure at Pass 2 exit	688.0 psia

D. DESIGN ANALYSIS

The following areas of the simulated shield form the basis of the structural analysis:

(1) Lifting Bracket. The lifting brackets have only one function--to aid in the handling of the replacement unit during shipping and installation. Because the brackets have no function during operation, they are subject to ground handling loads only.

(2) Support Flange. The shield support flange is subject to loads resulting from three separate conditions:

   (a) The difference in pressure between the dome end surface and the nozzle end surface.

   (b) The environmental loading imposed during ground testing.

   (c) Internal stresses induced by temperature distribution.

(3) Area Between the Flow Holes. The metal between the coolant flow holes in the aluminum simulated shield is subject to internal stresses caused by temperature gradients generated in the simulated shield during ground testing.

1. Lifting Brackets

There are three lifting brackets bolted to the dome end surface of the simulated shield. These brackets are used to handle the replacement unit when the reactor is in the nozzle-down position. They are mounted 120 degrees apart, with the center of each bracket located approximately 18 inches from the center of the reactor core axis. The base of each bracket is recessed into a pocket milled in the simulated shield to a depth equal to the bracket base thickness. The vertical web portion, which is hook shaped at one end, is designed to engage in a special lifting fixture.

The lifting fixture is designed so that a vertical load will be applied to each bracket. Since the brackets are mounted at an angle of 28° 30', with respect to the horizontal, the mounting bolts will be loaded in shear as well as tension (Figure 4-4).

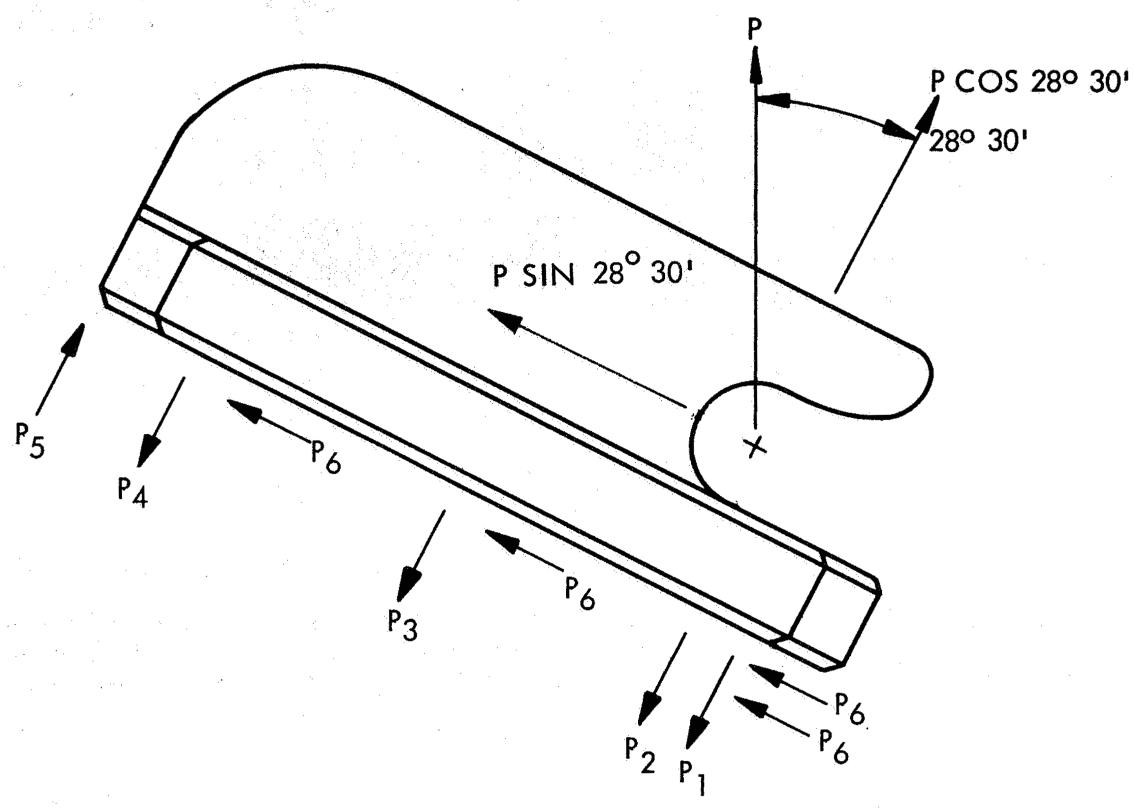

Figure 4-4. Lifting Bracket Applied Load Resolved Into Components

Attachment Bolts. The shear load on the attachment bolts is calculated using the 2-g axial acceleration load specified in WANL-TME-476. The shear component is P sin 28° 30', and the weight of the simulated shield is 2330 pounds.

The assumptions used to determine the tensile load in each bolt are:

(1) The vertical rib portion of the bracket is assumed infinitely rigid.

(2) The simulated shield is infinitely rigid.

(3) The bolt deflection is proportional to the bolt load.

(4) The influence coefficient at each bolt reaction point depends upon bolt elongation, flange bending, and flange shear.

The mounting bolts are 5/16-inch diameter, NF thread, and are made of 304 annealed stainless steel which has the following properties:

Tensile yield	30,000 psi (1600 pounds)
Shear yield	17,300 psi (1330 pounds)

Since the maximum shear load on the bolts is 183 pounds, as compared to an allowable yield shear load of 1330 pounds, the bolts are considered satisfactory in shear for the ground handling condition.

Since the maximum tensile load on the bolts is 695 pounds, as compared to an allowable tensile yield load of 1600 pounds, the bolts are considered satisfactory in tension for the ground handling condition. These values are based on a bolt preload which is approximately equal to the largest (695 pounds) tensile load in any bolt.

Threaded Inserts. The bracket mounting bolts are threaded into inserts installed in the simulated shield. This bolt can develop the full strength of a 5/16-inch diameter bolt which has a material ultimate tensile strength of 90,000 psi (4810 pounds). The maximum bolt load calculated previously is 695 pounds; therefore, the inserts are considered satisfactory for the 2-g ground handling condition.

Local Stress of Bracket at Bolt Attachment. The bolt washer face applies a uniformly distributed load (W) on the bottom of the counterbore in the bracket at the attachment locations (Figure 4-5).

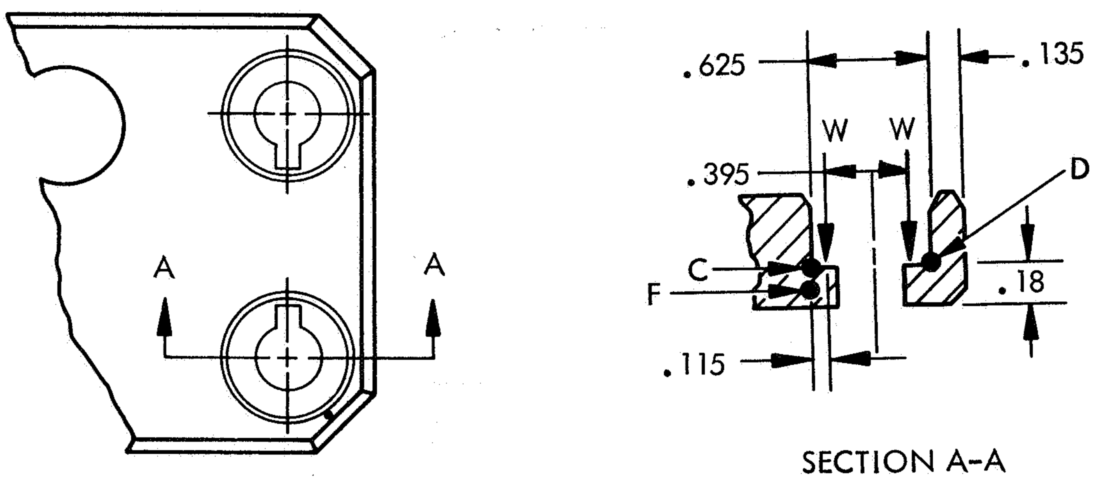

Figure 4-5. Section of Bracket at Bolt Attachment Holes

When computed bending stress, 15,800 psi, is compared to the tensile yield strength of 105,000 psi for Inconel X-750 at room temperature it is evident that the bottom of the counterbore in the bracket is satisfactory in bending for a 2-g ground handling load. Furthermore, a comparison of the shearing stress, 2410 psi, to this shearing yield strength of 60,500 psi for the material at room temperature, indicates that the bottom of the counterbore in the bracket is satisfactory in shear for a 2-g ground handling load. This shear stress will be increased as material is removed by the keyway slot; however, the effect of the reduction will be small and is neglected in this analysis.

Bending of the Bracket. Lifting and the bolt reaction loads, P_1 and P_2, cause the handling bracket to bend across Section A-A (Figure 4-6). In addition, Section A-A is only partially effective because of the flow holes in the bracket base.

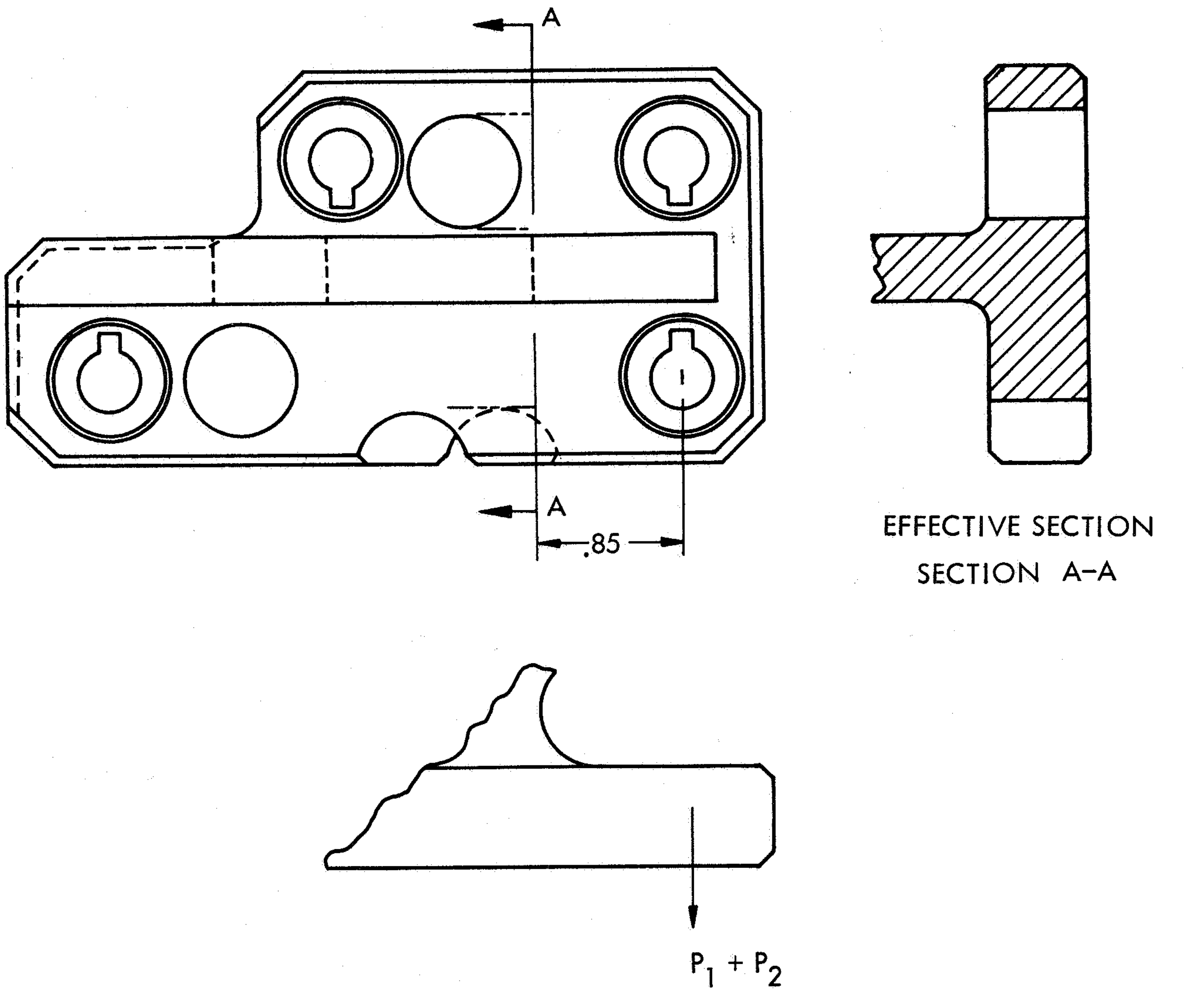

Figure 4-6. Bending of Bracket Base

Nevertheless, when this bending stress, 12,700 psi, is compared to the tensile yield strength of 105,000 psi for Inconel X-750 at room temperature, it can be seen that the base of the bracket is satisfactory in bending for a 2-g ground handling load.

<u>Lifting Lug</u>. In computing the strength of the lifting lug on axial load of 2-g is assumed (WANL-TME-476). The weight of the simulated shield is 2330 pounds. Then, selecting values for x and measuring the resulting h from Figure 4-7, values of σ are computed as shown in the following tabulation:

x	h	σ
0.04	0.632	2970
0.08	0.682	3600
0.12	0.700	5100
0.16	0.740	6500
0.20	0.792	9500
0.24	0.856	9780
0.28	0.940	9500

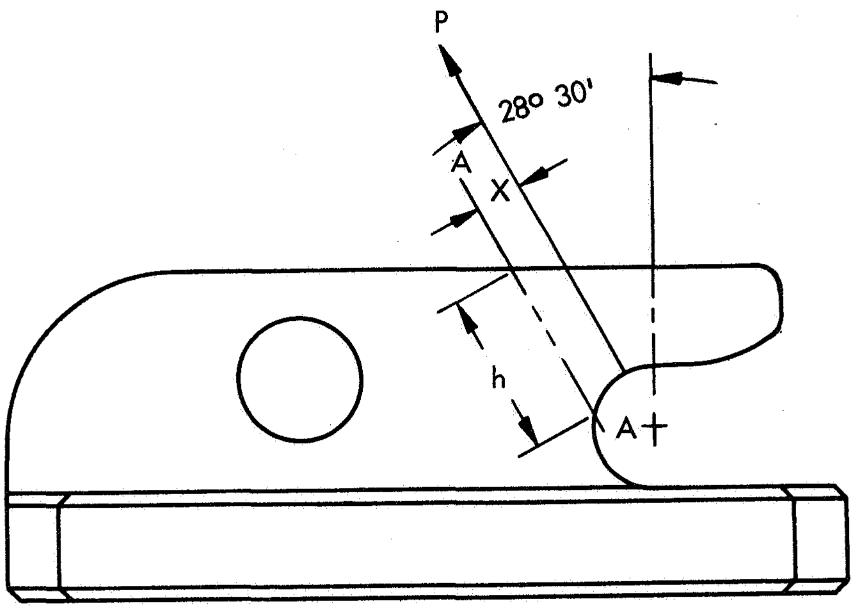

Figure 4-7. Lifting Lug

The maximum bending stress of 9780 psi, as compared to a tensile yield strength of 105,000 psi for Inconel X-750 at room temperature, indicates that the lifting lug is satisfactory in bending for a 2-g ground handling load.

2. Support Flange

The simulated shield support flange is subjected to loads from differential pressure, temperature distribution, and ground test inertia. Because the change in temperature between the support flange and the simulated shield is very small, the effect of the temperature distribution is neglected in this analysis.

The following assumptions are used in this analysis:

(1) The simulated shield is supported entirely by cantilever beam action of the flange.

(2) The clamping action of the pressure vessel prevents rotation of the flange.

(3) The load (W) is reacted at the point shown in Figure 4-8; this point corresponds to the mid-point of the bearing area.

(4) The reflector support flange does not contribute to the strength of the simulated shield support flange.

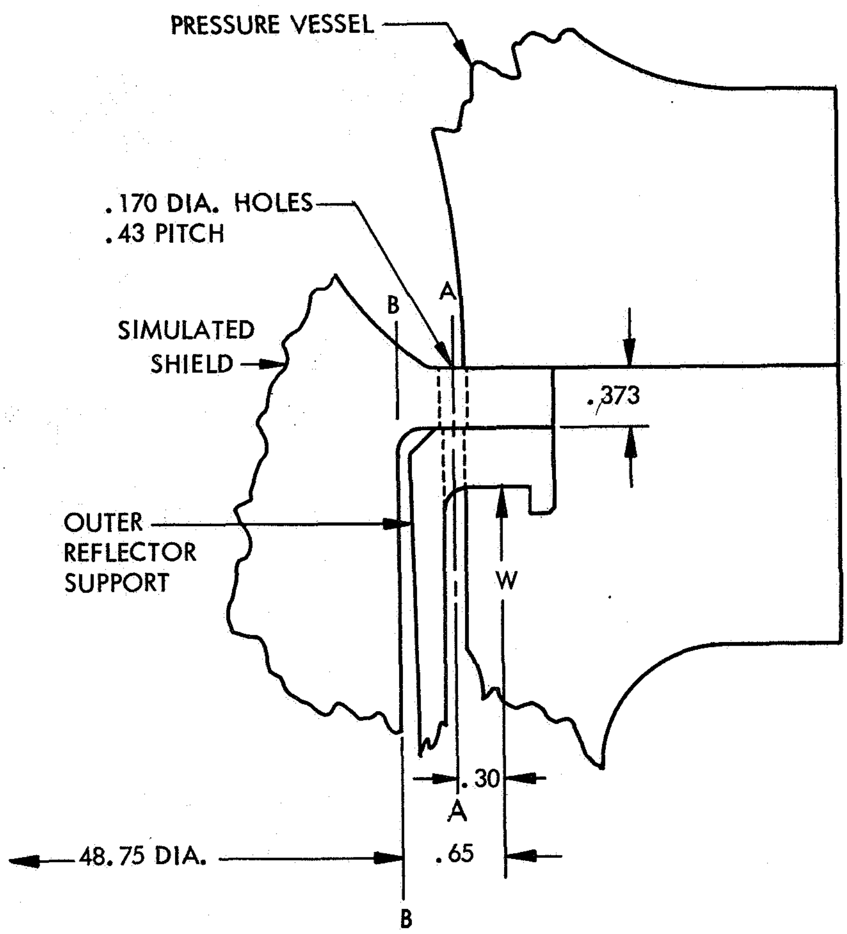

Figure 4-8 Support Flange

Stresses Caused by Axial Loads. The maximum engine ground test inertia load (14-g axial acceleration) is specified in WANL-TME-476. The simulated shield weight is 2330 pounds; therefore, since the inertia requirement is 14-g, the flange must support 32,620 pounds.

The bending stress of 9250 psi, as compared to a material yield strength of 36,000 psi for 2219-T852 aluminum, indicates that the simulated shield support flange is satisfactory for the 14-g transient inertial load specified for engine ground tests. Comparison of the 1010 psi shear stress against the 24,000 psi material yield strength indicates that the flange is satisfactory for the 14-g transient inertia load specified for engine ground tests.

3. Thermal Stresses Due to Radial Temperature Distributions Around Flow Holes

The area between the flow holes in a direction normal to the reactor axis is subjected to internal loads caused by the temperature distributions. The maximum thermal stress will occur at the surface of the flow holes near the center and at the aft end of the simulated shield. Since the walls of the flow holes are cooler than the metal between them, the tangential stress at the surface of the holes will be in tension, changing to compression between the holes. The structure around the flow holes is idealized as a thick-walled cylinder as shown in Figure 4-9.

The 9750 psi stress, when compared to a material yield strength of 27,000 psi (the axial direction) at 300 to 350° R for 2219-T856 aluminum, indicates that the simulated shield will perform satisfactorily in the area of the flow holes during nuclear ground testing.

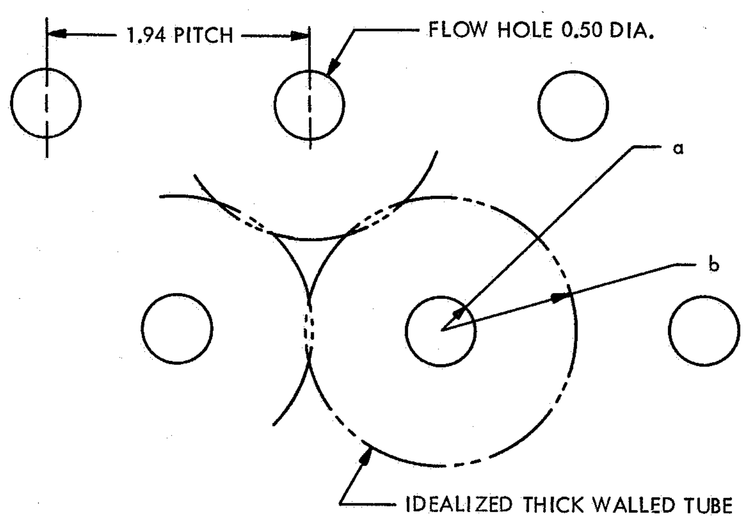

Figure 4-9. Shield Section Showing Flow Holes

E. VIBRATION ANALYSIS

The natural frequency of the simulated shield in the axial direction was determined analytically; the following assumptions were made for this analysis:

(1) The simulated shield support flange acts as a simple cantilever beam between the flow holes; the beam is fixed in the pressure vessel closure flange and has a concentrated load on the opposite end.

(2) The simulated shield support flange does not rotate in the pressure vessel closure flange.

(3) The reflector support cylinder, which is also trapped in the pressure vessel's closure flange, does not contribute to the strength of the simulated shield support flange.

These assumptions, along with a factor for ring restraint $(1 - \nu^2)$, which is Poisson's ratio for aluminum, result in a conservative analysis.

The load (l) on the support flange is a function of simulated shield weight (2330 pounds), the circumference of the flange, and the material removed from the flange by the coolant flow holes.

The deflection of the beam is determined from the P, and the moment of inertia of the beam. Substituting the values in the equation,

$$f = \frac{1}{2} \sigma g$$

Where g is the gravitational constant, 386 in./sec.^{2}, σ is the deflection of the support flange under a 1-g load. The natural frequency (f) of the replacement unit is 858 cps.

This frequency is higher than any expected during the ground handling and assembly conditions. It is not, however, beyond the frequency spectrum for transient and steady-state conditions. The support flange analysis indicates that the flange is satisfactory even if it were excited at the 14-g level.

One of the prime consideration throughout the design of the Block I shield was the precautions taken to minimize the possibility of the instrumentation leads vibrating excessively during the ground handling and engine operating conditions. Consequently, all leads are clamped to the unit, and the exposed ends are as short as possible.

Calculations indicate that a spacing of 2.10 inches between support clamps for thermocouple sheaths would ensure a natural frequency in excess of 2000 cps. Since thermocouples and other instrumentation leads are clamped every 4 inches against the surface of the simulated shield, the natural frequency is below that level. In addition, the damping introduced by clamping against the surface tends to limit the amplitude of vibration.

The ultimate strength of Type-304 stainless steel, the sheath material, at an arbitrary temperature of 70°F is 62,000 psi. The endurance strength of the material is estimated as:

$$\sigma_b = 0.45\ (62,000) = 27,900 \text{ psi.}$$

This calculation indicates that the material is capable of 10^6 bending cycles at a bending stress of 27,900 psi before failure occurs. Comparison of the thermocouple sheath bending stress (0.1744 psi) with the 27,900 psi endurance limit of the material, indicates that the instrumentation leads in the simulated shield will not fail during ground testing.

V. FLOW SCREEN

Design Philosophy. The flow screen, which is located between the shield and core support plate at the core inlet, serves to filter the coolant flow to make certain that the fuel orifices are not clogged by foreign particles, a condition which could cause over temperaturing in the fuel elements and structural failure in the core.

Description. The screen is mounted on the inlet end of the core support plate between the core and the shield replacement unit. It is elliptical in shape, with the convex side toward the shield replacement unit (Figure 5-1). The structural support part of the screen assembly is made of 1/8-inch stainless steel. The required coolant flow area is supplied by 2370 holes (7/16 inch in diameter) drilled through the support structure on a triangular pattern and an additional 249 holes (5/16 inch in diameter) drilled near the outer periphery. A 0.007-inch diameter, sintered, stainless steel mesh screen is brazed to the convex side of the support structure. The space between wires in the screen is 0.007 of an inch, which is 1/3 the size of the smallest flow orifice (0.021 inch) used in the core fuel elements.

Material Selection. Type 304 stainless steel was chosen for the structural member of the flow screen because of its strength and ductility an cryogenic temperatures. The mesh is made of Type 316 sintered stainless steel wire. This sintered screen is preferable because it does not separate when the mesh is rolled over the support structure, and the openings between wires remains uniform over the entire assembly.

Design Analysis. The screen support structure was analyzed as a shell with a uniform pressure load on the convex side; it was assumed that the screen did not contribute to the mechanical strength of the support structure. The maximum anticipated pressure drop through the screen is 4 psi. The resulting stresses and deflections derived from the analysis were low, indicating an ample margin of safety and it should also be noted that the margin of safety against buckling is also high. Samples of the brazements between the screen and support structure were examined both visually and with photomicrographs to determine the quality of the bond between the stainless steel wires of the mesh and the depth of penetration of the braze metal. The photomicrographs show that there is still bonding between the wires, even after the mesh is peeled from the experimental support plate; however, the bonds are not strong and, therefore, it is assumed that the mesh does not contribute to the strength of the support structure. It was also determined that the brazing operation had no adverse effects on the sintered bonds between wires. The depth of penetration of the braze metal was approximately 0.0002 inch.

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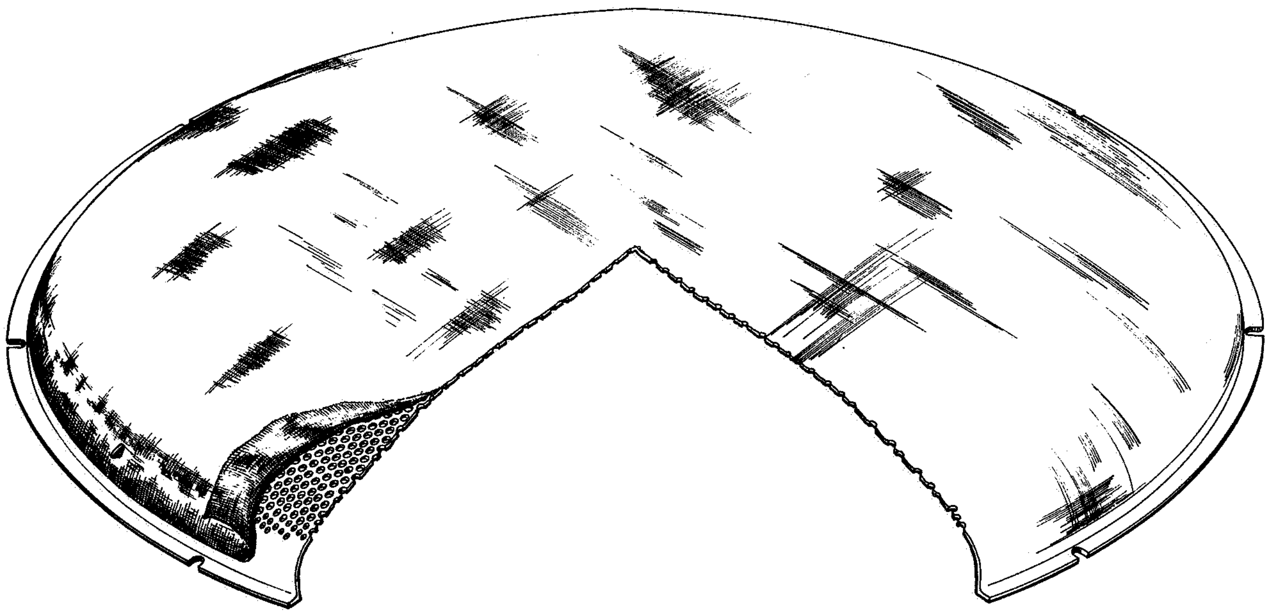

FIGURE 5-1. FLOW SCREEN

VI. OUTER REFLECTOR

The 52-inch long NRX-A beryllium reflector is a hollow cylinder with an O.D. of 49.369 inches and an I.D. of 40.131 (Figure 6-1). While it is in many ways similar to the one used in the KIWI-B reactors, it differs in that it is formed into a continuous cylindrical structure as opposed to the 12 discreet sectors which are fastened separately to the pressure vessel in the KIWI design. The NRX-A cylinder is comprised of twelve 30-degree sectors, and the cylinder formed from these sectors is clamped between the two titanium rings by 36 titanium tie bolts. The assembled cylinder contains twelve 4-inch holes, one at the center of each of the discreet sectors; these holes provide the room needed to insert the control drums. Each control drum, in turn, is supported by two bearings which are held by bearing housings at the support rings. Cooling passages in the beryllium reflector are provided by three-hundred-and-eighty-four 0.185-inch diameter axial holes which penetrate the sectors. Pilot fits at each end of the drum cavity are used to locate the sectors with respect to the titanium support rings. Locating the sectors in this manner provides maximum control over the nominal 0.050-inch radial clearance between drum and reflector sectors. In addition to the required coolant channels, each sector contains 14 shim rod holes. Aluminum or beryllium shim rods, 0.5 inch in diameter, can be inserted in these holes to control neutron leakage from the reactor.

The outer reflector assembly is supported by the z-shaped dome end support ring which is positioned at one end between the pressure vessel and the pressure vessel closure flange. Axial tie bolts are used to secure the support ring to the reflector segments near its outer diameter.

Each control drum consists of a beryllium cylinder approximately 4 inches in diameter and 52 inches long. This cylinder has axial holes which provide channels for coolant flow; the face of the cylinder has a 120-degree, segmented poison plate. The poison plate is 0.050 inch thick and is fabricated from an alloy consisting of elemental boron (enriched in the B^{10} isotope) dispersed in an aluminum matrix. The assembly is enclosed in an aluminum sleeve to make certain that the drum is not jammed in the unlikely event that pieces break off the poison plate or the beryllium cylinder. Each end of the sleeve is welded to an aluminum flange which has an extension that serves as a bearing shaft. A coil spring between the aft end flange and the beryllium cylinder absorbs the differential thermal expansion between the beryllium and aluminum.

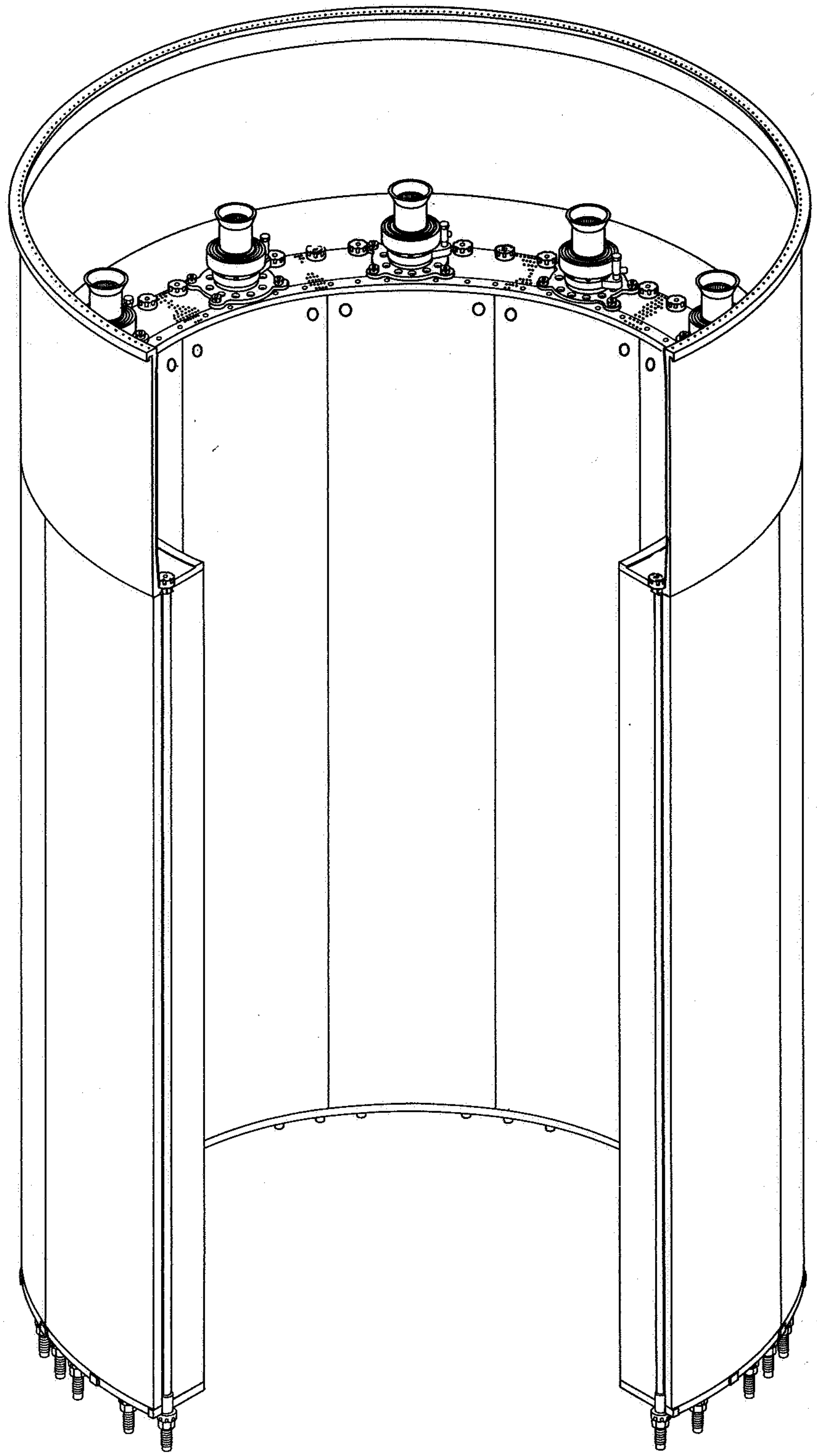

FIGURE 6-1. OUTER REFLECTOR ASSEMBLY

The control drum is supported by two ball bearings located in their bearing housings at the support rings. While the radial loads are carried by both bearings, all thrust loads are carried by the dome end bearing. The bearing housing at the aft end is an integral part of the nozzle end support ring, while the bearing housing at the dome end is bolted to the dome end reflector support ring.

Control drum position is regulated by individual drive mechansims connected through drive shafts which penetrate the pressure vessel dome. Connection between the drive shaft and the drum coupling is achieved with a spline. A mechanical stop is provided to prevent the drum from rotating beyond its full-in or full-out position. A locking pin can be inserted through this stop into the bearing housing at the full-in position to prevent any drum motion when such protection is desired. A Type 304 stainless steel torsion spring serves to keep the drum in the full-in position when a driving force is not applied to the shaft; this spring has a torque of 16 to 24 in.-lb.

Essentially, the outer reflector assembly has two functions: one nuclear, the other mechanical. In a nuclear sense, it reflects the neutrons back into the core in order to keep the chain-reaction process going. It also serves to house the control drums which control the reflectivity of the assembly, thereby keeping the core at the proper level of operation. In a mechanical sense, the assembly supports the entire reactor in the pressure vessel; it supports the core by the core support ring, which, in turn, is attached to the core support plate. This whole assembly is then supported in the pressure vessel by the dome end support ring.

In an environmental sense, the outer reflector assembly is subjected to a mixture of gaseous and liquid hydrogen during high neutron and gamma irradiation. The hydrogen enters the outer reflector at 162° R and 750 psia and leaves at approximately 240° R and 700 psia. Details of the fluid flow and thermal analysis can be found in WANL-TNR-128, Vol. III, while details of the nuclear and radiation analysis can be found in Vols. I and II, respectively. In addition to the thermal and nuclear environments the outer reflector assembly is subjected to shock and vibration which is defined in WANL-TME-476. The writeups of the various components which follow give a brief description of the environment; however, detailed information should be derived from the reports cited.

A. DOME AND NOZZLE END SUPPORT RINGS

Design Philosophy. The dome end support ring (Figure 6-2) supports the outer reflector assembly in the pressure vessel. It also supports the core of the reactor via the core support ring. The design of the dome end support ring is interdependent with many other design decisions made on the NRX-A reactor. For example, the decision to install control drums through the ring, using the bearing housings to maintain alignment between ring and sectors, makes it necessary to drill large holes through the ring. The size of these holes reduces the space left for metal, and hence makes a monimal stress capability mandatory. The fact that the core is supported through this ring also places a severe demand on it. The ring must not only withstand the core loads, but must also be flexible enough to prevent the cone-shaped transition member between the core support plate and the ring from being overstressed, particularly in view of the fact that the support plate shrinks approximately 0.0625 inch radially at operating conditions. In view of all of these inter-related conditions, the accent has been placed on developing a balanced design in which the demands made on all components are essentially equal. The decision to have an outer reflector assembly which could be handled as a unit also did much to dictate the design of the dome end support ring.

The reflector assembly is centered in the pressure vessel by a double spigot fit at the pressure vessel flange area. The core support ring is held by 60 bolts, 36 of which are bolted directly to the dome end support ring at the inner diameter while 24 penetrate the dome end support ring and a portion of the sectors near the inside diameter of the outer reflector. At this point, the bolts are fastened to threaded bushings which fit into pockets machined out of the beryllium sectors.

The nozzle end support ring, in turn, was designed to align and support the sectors in the outer reflector assembly (Figure 6-3). This supporting function is accomplished in conjunction with the tie bolts and the dome end support ring; the sectors are clamped between the two rings by the 36 tie bolts. The sectors are aligned by means of a close-tolerance spigot which fits into the nozzle end support ring. All shim rods are retained at the nozzle end ring by the shim rod nut which is threaded into the ring. By retaining the shim rods in this way, it is possible to interchange them without disturbing other parts.

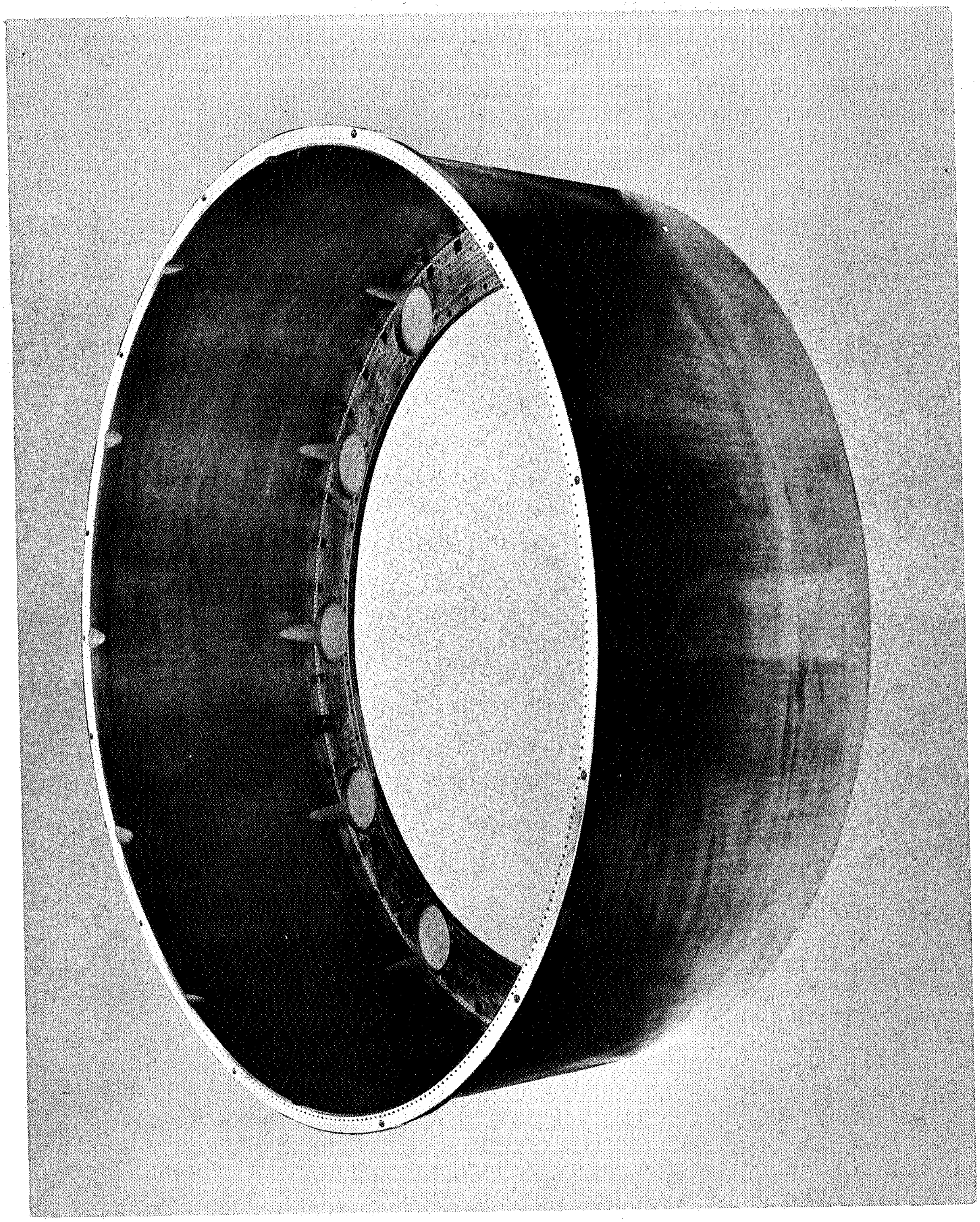

FIGURE 6-2. DOME END SUPPORT RING

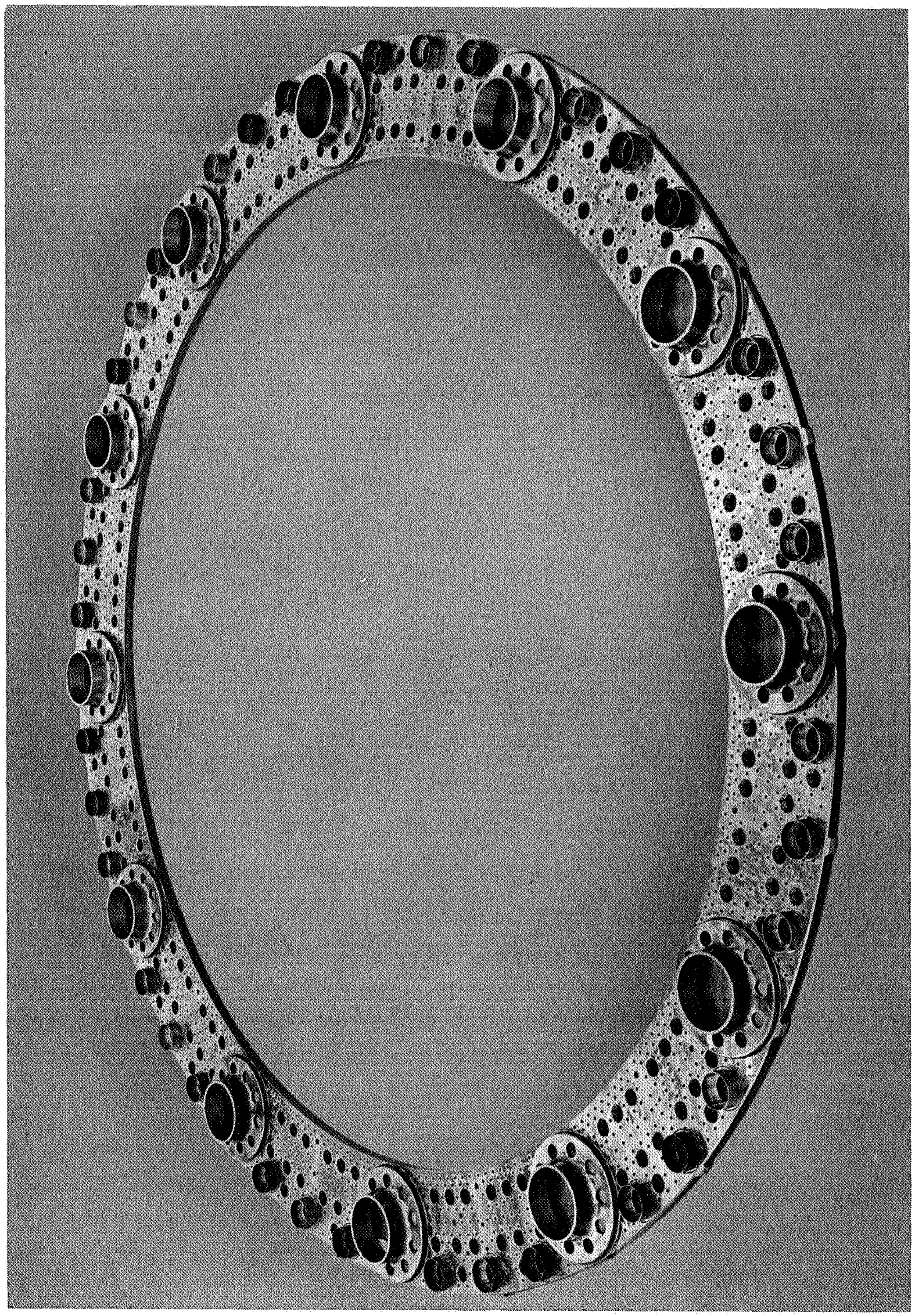

FIGURE 6-3. NOZZLE END SUPPORT RING ASSEMBLY

Description. The dome end support ring is approximately 0.5 inch thick, having an I.D. of 40.131 inches and an O.D. of 49.369 inches. Twelve 4-inch control drum, 36 tie bolt, and approximately 3650 coolant holes penetrate the ring. Plenums between the ring and the sectors connect the coolant passages. These plenums are necessary since many more passages are required for the ring than the sectors. The ring has a thin cylindrical extension at the O.D. which extends forward about 16 inches. This cylinder terminates in a small external flange 0.380 inch thick by 0.75 inch wide. The flange is trapped between the forward flange of the pressure vessel to support the reflector in the vessel. Three hundred and sixty 0.113-inch diameter holes penetrate this flange and act as an orifice for the coolant passing between the pressure vessel and the O.D. of the outer reflector. At the O.D. of the small flange, there is a lip on the aft side. This lip, which is 0.182 inch long by 0.142 inch wide, fits into a groove in the pressure vessel flange to center the reflector in the vessel. In addition to the lip, there is a locating pin which orients the reflector in the pressure vessel and aligns the drive shaft ports.

The nozzle end support ring is a continuous ring with 12 integral bearing housings that support the nozzle end control drum bearing. The ring supports the sectors and clamps them to the upper support ring. Projections, approximately 1/2 inch long located on the centerline of each control drum and between each control drum, space the reflector relative to the pressure vessel to assure a uniform coolant flow path.

In addition to the 12 holes for the control drum bearings, there are 168 tapped shim rod holes, 36 tie bolt holes and approximately 1600 coolant holes. Like the dome end ring, the coolant holes are connected to the sectors by a plenum.

Material Selection. The selection of a material for the NRX-A reflector dome end support ring and nozzle end support ring is based upon several interacting mechanical considerations: adequate strength is necessary; a low thermal expansion coefficient reduces strength requirements and improves mechanical fit-up with the reflector; similarly, a low elastic modulus reduces strength requirements.

Among the materials considered were 9% nickel steel, austenitic stainless steels, aluminum alloys, titanium, Inconel X-750, and Inconel 600. The 9% nickel material was eliminated because it loses ductility at operating temperatures, the austenitic stainless because of its high thermal expansivity and high stresses, and Inconel 600 because of high stresses, Inconel X-750 was eliminated because it exhibited marginal notch-sensitivity in cryogenic applications and, finally, aluminum was dropped because of its poor expansivity properties and relatively low strength. Only the 5% aluminum, 2-1/2% tin-Titanium alloy exhibited all the desirable characteristics delineated above, and was also deemed metallurgically adequate.

In addition, the extra low interstitial 5% aluminum, 2-1/2% tin-Titanium alloy not only maintained excellent strength over the operating temperature range, but also provided the added advantages of superior ductility and notched tensile strength when the iron content was properly controlled. A detailed review of the metallurgical characteristics and the limited irradiation effects data for this alloy indicated that no significant effects on properties were observed as a result of the NRX-A1 operation. Since the alloy provides adequate margin on reliability, it was selected for the support rings.

Design Analysis. Extensive analyses were performed on two elements of the dome end support ring: the flange and the plate.

(1) Flange. At full-power, steady-state operation the flange is conservatively estimated to contract 0.035 inch and rotate inward 0.0126 radians due to the combination of the pressure vessel contraction and the reactor fluid pressure drop. The stress distribution in the cylinder for this condition is shown in Figure 6-4. The peak equivalent stress is 25,200 psi, a figure which compares favorably to the allowable stress (0.80 ultimate) of the 5A1-2.5 sn-titanium, which is 1000,000 psi at the design operating temperature of 300° R.

Stresses in the flange have also been investigated for a conservative deflection of 0.033 inch outward and 0.020 radians inward rotation. These values are equivalent to those obtained for a cylinder 300° R cooler than the pressure vessel which is assumed to be at room temperature. For this condition, which is more severe than any expected transient conditions, the peak equivalent stress is 83,000 psi which is still less than the allowable 100,000 psi.

In addition, analysis of the stability of the thin-wall cylinder for 4-g axial and transverse loads indicates that the design is completely satisfactory in this respect with a margin of safety over three, based on conservative empirical equations.

(2) Plate. The complexity of the load distribution, temperature, and geometry involved makes the use of a number of simplifying and conservative assumptions necessary.

Referring to Figure 6-5, the stresses resulting from the steady-state mechanical loads are as follows: the local bending stress under the tie bolt head "a" is less than 60,000 psi, the corresponding bending plus membrane stress at "b" is less than 70,000 psi, and the hoop stress at the control drum cut-out "c" is less than 35,000 psi. The full-power,

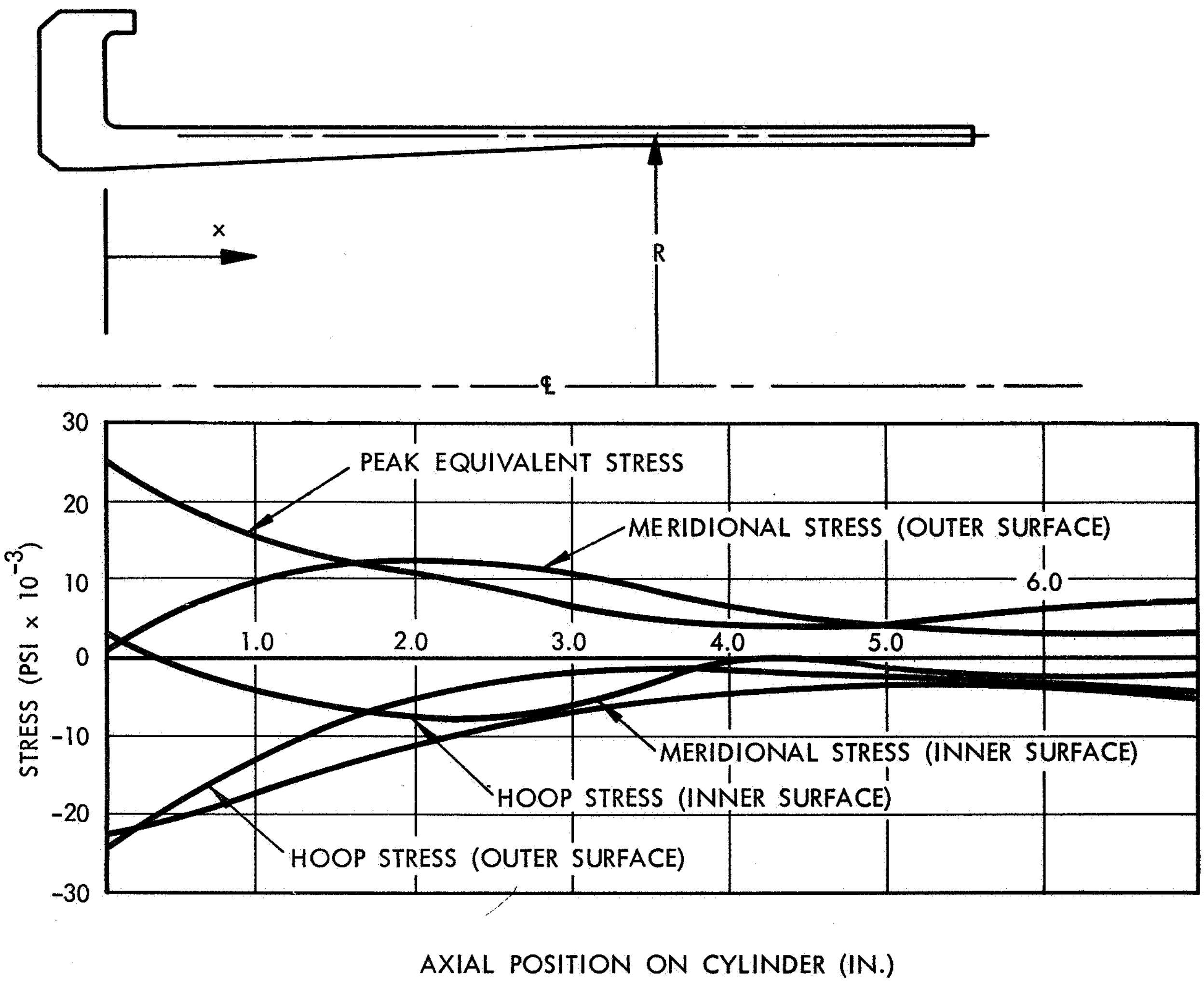

FIGURE 6-4. STEADY-STATE STRESS DISTRIBUTION

steady-state temperatures at points "a" and "b" are less than 500° R. Conservatively assuming titanium material properties at 450° R, the allowable stress based on 80% of ultimate is 85,000 psi.

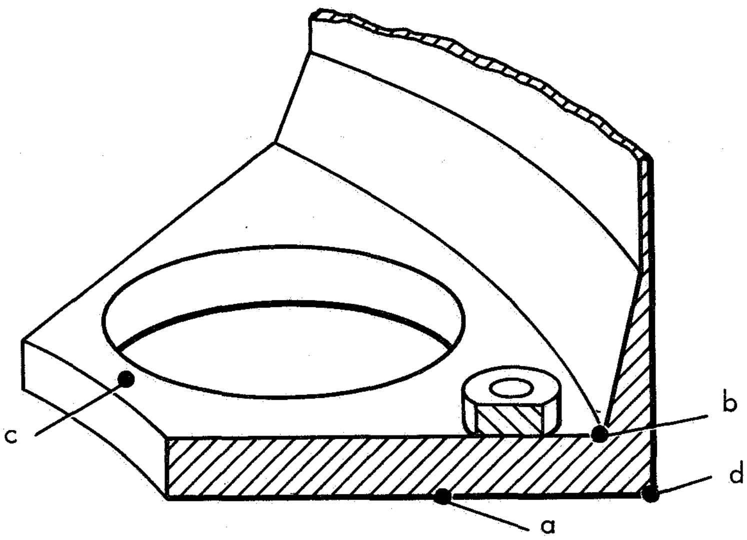

Figure 6-5. Dome End Support Ring Plate

The thermal stresses caused by radial and circumferential temperature gradients in the plate are less than 4000 psi. There is a possiblity that the inside surface diameter of the plate might be relatively hot due to poor cooling. This condition could result in a local hoop compressive stress of 35,000 psi at the inside surface of the plate. Because of a 200° R axial temperature gradient, resulting from greater cooling on the surface between the plate and beryllium reflector sector, the radial and tangential stresses at the colder surface could reach 8000 psi, providing the plate does not bow. Because the titanium plate is operating at a higher temperature than the beryllium sector and because the plate is bolted to the beryllium, there will be compressive stresses in the plane of the plate. If the plate does not slip, the stress could reach 20,000 psi. However, even with a coefficient of friction as high as 0.5, slipping does occur, reducing the stress to 10,000 psi compressive. Since all of the thermal stresses are small, when they are superimposed on the mechanical stresses and considered in a fatigue analysis, they are acceptable because of the low number of cycles involved.

Based on conservative calculations, the gap between the plate at point "d" and the beryllium reflector sector could be as large as 0.004 inch; this gap results from the mechanical load. The tendency of the plate to bow due to the axial temperature gradient reduces this existing gap by 0.003 inch. Based on a conservative estimate, the plate could also separate by as much as 0.002 inch from the sectors at the center of the radial joint between sectors. In either case, there should be little redistribution of coolant flow due to separation of the support plate from the beryllium reflector.

The nozzle end support ring contains the nozzle end control drum bearing. It also serves to restrain the radial motion of the nozzle end of the beryllium reflector sectors. This radial deflection is caused by the core loads transmitted to the dome end support ring and the radial pressure drop across the sectors.

The average radial deflection of the nozzle end support ring caused by the mechanical loads is 0.0055 inch inward. If it is assumed that the beryllium sectors do not stiffen the ring at the bearing housing location, a conservative value for the stress in the ring at the bearing housing is less than 10,000 psi compressive. Superimposing the stress caused by the axial and radial thermal gradients, the maximum stress is 18,000 psi compressive at the mean diameter on the nozzle end face. As previously shown, this is a low stress for titanium.

Environment. During steady-state conditions gaseous hydrogen at a pressure of approximately 730 psi and a temperature of 220° R flows past the dome end support ring. A mixture of gaseous and liquid hydrogen flows over the nozzle end support ring at a temperature of 162° R and a pressure of 750 psi during steady-state conditions.

During start-up, both rings could be subjected to a mixture of gaseous and liquid hydrogen at temperatures of 162° R.

B. SECTORS

Design Philosophy. A hollow beryllium cylinder surrounds the core and acts as a neutron reflector. The reactivity of the core is controlled by 12 control drums which are equally spaced in this beryllium cylinder (Figure 6-6). Because relative motion between the titanium support rings and the beryllium cylinder could bind these control drums and cause the cylinder to be over-stressed, it is advantageous to break the cylinder up into twelve 30-degree sectors with a control drum at the center of each sector. In this configuration, each sector acts independently of the others, the fixed point being the centerline of the control drum hole.

The sectors are located with respect to the support rings by pilot fits at the control drums holes. At the dome end support ring, the bearing housing pilots through the ring and into a counterbore in the sector. The fit in the ring and the sector is controlled to 0.0005 inch. At the aft end, a spigot on the end of the sector pilots into a counterbore in the nozzle end support ring, and the fit between the sectors and the ring is controlled to 0.0005 inch.

The gap between sectors varies from 0.011 to 0.044 inch, depending on tolerance build-up. The gap prevents binding in the event that the rings shrink faster than the sectors; a situation which occurs during transient conditions only. Normally, the rings are at a higher temperature than the sectors, thereby increasing the gap. The shim rod locking devices and the tie bolt bushings are designed so that they do not interfere with the motion of the sectors.

In addition to the control drum hole, there are fourteen 0.500-inch shim rod holes, thirty-two 0.187-inch coolant holes, and two 0.688-inch tie bolt holes in each sector.

Description. Twelve sectors make up a hollow cylinder to form the outer reflector. Each 30-degree sector is 52 inches long with a 4-inch hole at its center. Several coolant passages, spaced to provide uniform temperature distribution, penetrate the sectors. In addition to the coolant passages, there are fourteen 0.5-inch diameter holes which allow for nuclear shimming. Shimming is accomplished by substituting aluminum for beryllium rods. The sectors are clamped between the two titanium support rings with 36 preloaded titanium tie bolts to prevent separation. In addition to the tie bolts, 24 titanium bolts tie the inner edge of the dome end support ring to the sectors. Tapped Ni Span-C plugs are inserted in the sectors so that the beryllium will not have to be tapped for these bolts.

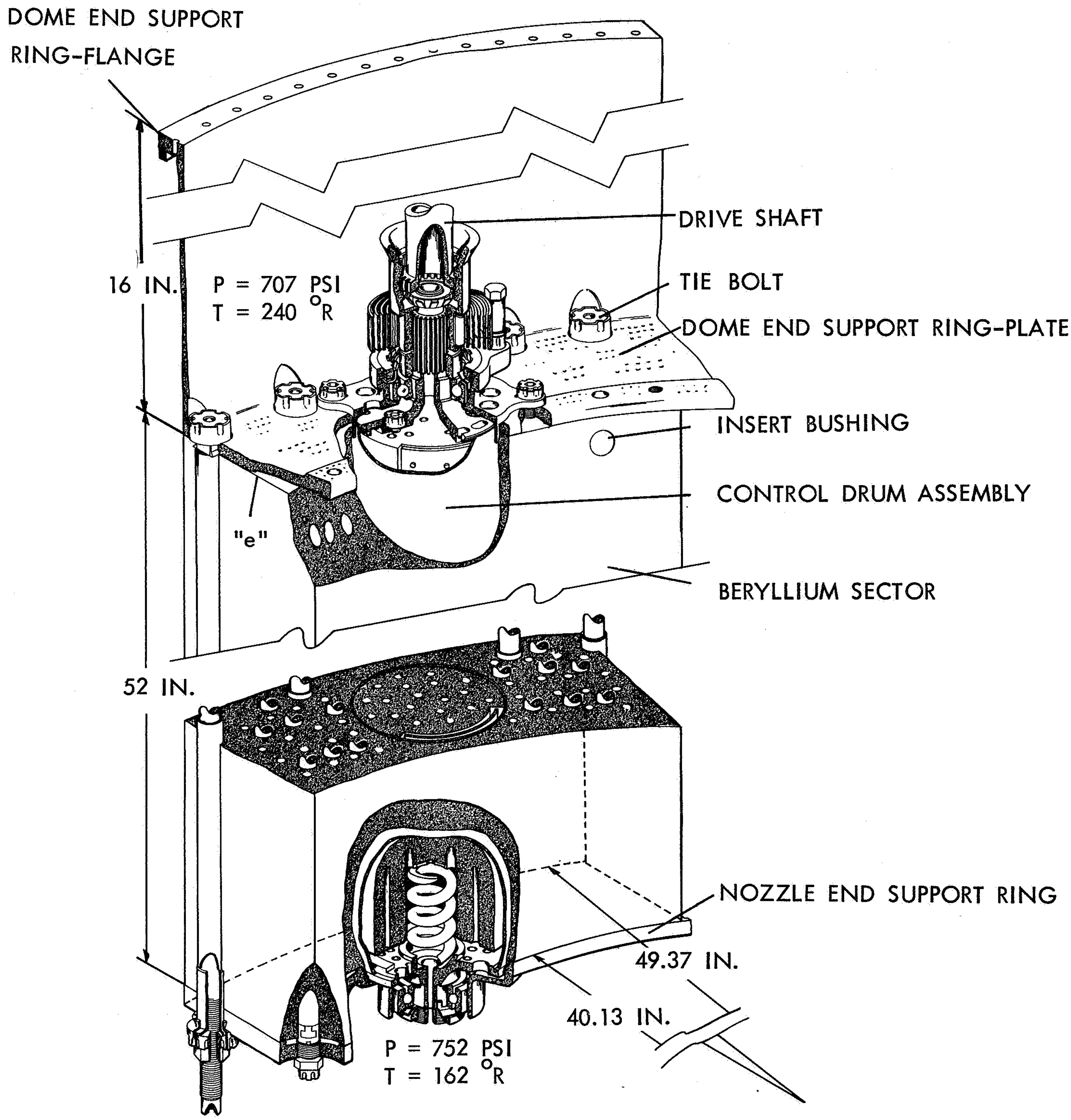

FIGURE 6-6. TYPICAL SECTOR

The sectors are located with respect to the end ring by pilot fits at each end of the control drum hole. This arrangement provides the best means of centering the control drum. In addition to the pilot fits at the control drum holes, the sectors are located by bushing at each tie bolt. These bushings fit tight in the nozzle end support ring. They are approximately 0.004-inch loose in the sectors, thereby limiting rotation of the sectors, but not restraining them for thermal movement.

Material Selection. The sectors are made from vacuum-sintered beryllium because of its low neutron absorption cross-section, high neutron scattering cross-section, and light weight. The only material decision to be made was whether or not a high purity grade of beryllium should be used to facilitate re-use. Based on anticipated design change, it was decided that it would not be practical to consider re-use for the Block I reactors; however, high purity beryllium will be considered for later reactors.

Design Analysis. The maximum steady-state, full-power tensile stress due to the tie bolt preload, mechanical loads and thermal gradients is less than 10,000 psi, which is well within the yield stress of 27,000 psi.

The thermal stresses around the coolant holes are the most significant stresses in the sector. Considering the cooling hole with the largest temperature gradient and simplifying the problem to that of a radial temperature through a long cylinder, the maximum stress is 13,000 psi at the inside surface of the coolant hole. However, since the control drum hole is located in an area that is already in compression, the resulting stress is less than 10,000 psi.

Low ductility coupled with possible surface imperfections could result in a brittle fracture. The fracture toughness, which was determined experimentally at 140° R, was set at 1.0 in.-lb./in.^{2}. The elastic energy potential of the maximum bi-axial tensile stress state of 10,000 psi at the channel surface was calculated for axially and circumferentially oriented cracks. It was found that, for a surface with shallow cracks, the elastic energy potential was about 0.2 in.-lb./in.^{2}. This is lower than the minimum measure value of 1.0 in.-lb./in.^{2} required to propagate such a crack.

Insert bushings made of Ni Span-C are loacted at the inside diameter near the dome end in order to avoid tapping directly into the beryllium. The analysis shows that maximum bearing stress between the Ni Span-C bushing and beryllium is an acceptable 50,000 psi; this stress is caused by the bolt load. Since the titanium bolts are 1.2 inches long, they are able to accommodate the small lateral deflections caused by difference in thermal contraction between the titanium plate and beryllium sectors without yielding. The combined maximum axial and bending stresses in the bolt are 75,000 psi, while the allowable stress, based on 80% of the ultimate strength at design temperature, is 88,000 psi.

Environment. The sectors are subjected to gaseous and liquid hydrogen at temperatures ranging from 162° R to 230° R and pressures up to 750 psi. Because of the pressure differences between the outer reflector-pressure vessel annulus and the inner reflector-outer reflector annulus, there is slight pressure difference (average 15 psi) across the sectors. The outer reflector pressure vessel annulus experiences the higher pressure, thereby tending to force the sectors toward the center of the core.

C. TIE BOLT

Design Philosophy. The tie bolts (Figure 6-7) clamp the sectors between the dome and the nozzle end support rings. There are 36 tie bolts, 12 of which are located at the split between sectors, while the other 24 are equally spaced on either side of the 12 control drums. All tie bolts are located at the same radius, which is near the O.D. of the sectors. Locating the tie bolts at this radius reduces bending stresses in the dome end support ring where the axial load is transmitted to the thin-cylinder section. These bolts are preloaded to prevent separation of the rings and sectors. The preload is applied hydraulically at assembly so that the bolts will not turn and be damaged under load. The threaded end of the bolts is extended so that threads are provided to attach the hydraulic tensioning device. A square section under the head of the tie bolt fits into the dome end support ring to prevent the tie bolts from rotating while nuts are being tightened or loosened. The tie bolt is cooled by passing coolant through a hole in the center of the bolt. The threaded end of each tie bolt passes through a stepped bushing (Figure 6-8) which fits tightly in the nozzle end support ring. The bushing is machined to provide a controlled fit in the sectors. These bushings limit the rotation of the sectors about the control drum locating fits, yet allow the sectors to move thermally with respect to the nozzle and dome end support rings.

The tie bolt nut traps the stepped bushing in place and keeps the preload on the tie bolt. Both the tie bolt nut and the tie bolt head are locked with fail-safe locking devices. If a tie bolt breaks, the locking device will retain it and keep it from interfering with the operation of other components.

The 12 tie bolts located at the split between sectors are surrounded by a C-shaped aluminum sleeve which acts as a spring seal to limit coolant leakage from the I.D. to the O.D. of the reflector. This control is necessary if effective orificing is to be maintained for the two annuli bounded by the outer reflector.

Material Selection. Titanium was chosen for the tie bolt material because its thermal expansion characteristics are similar to beryllium and this material "compatibility" reduces thermal stresses which would be caused by differences in the expansion rate of disimilar materials. Titanium's high strength and light weight, combined with a low modulus of elasticity, also makes it a desirable tie bolt material.

Design Analysis. The initial preload placed on each tie bolt is 10,000 pounds, and the only time the tie bolt experiences additional load is during a transient condition

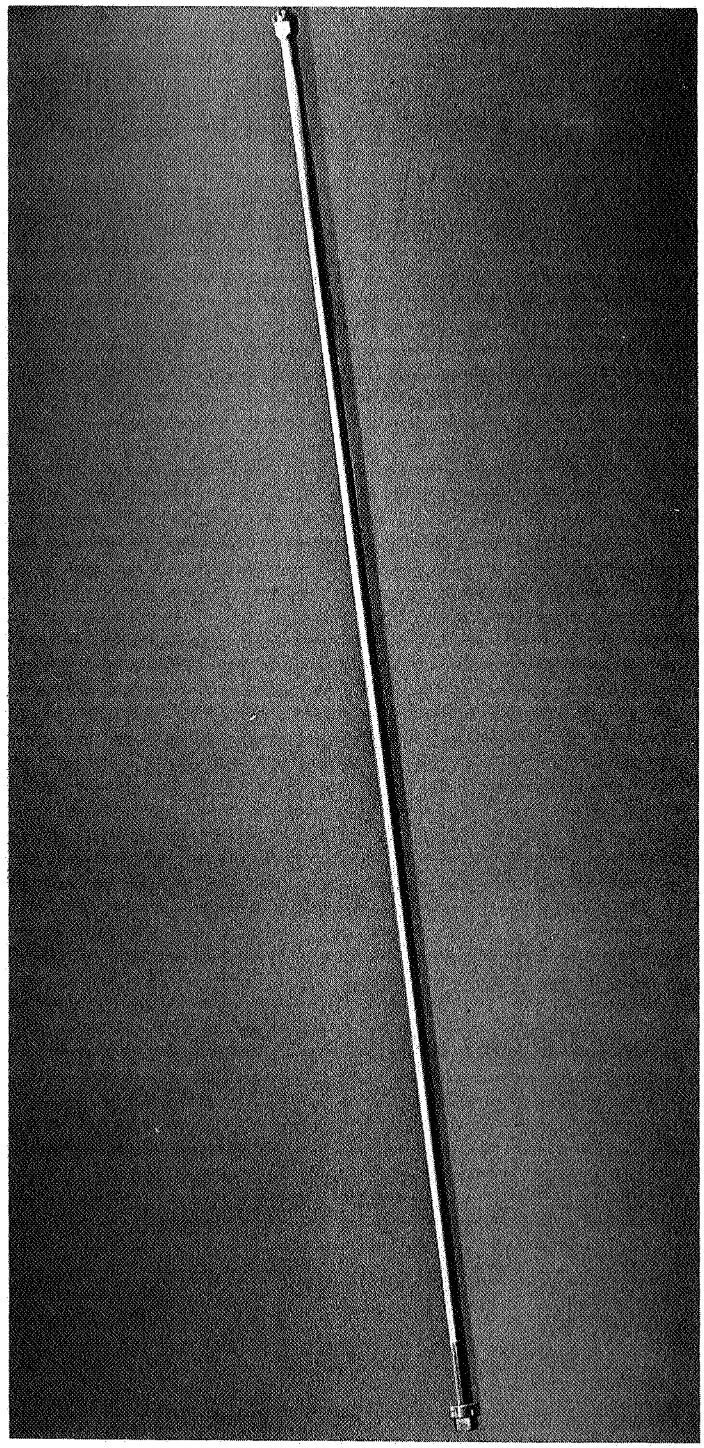

FIGURE 6-7. TIE BOLT

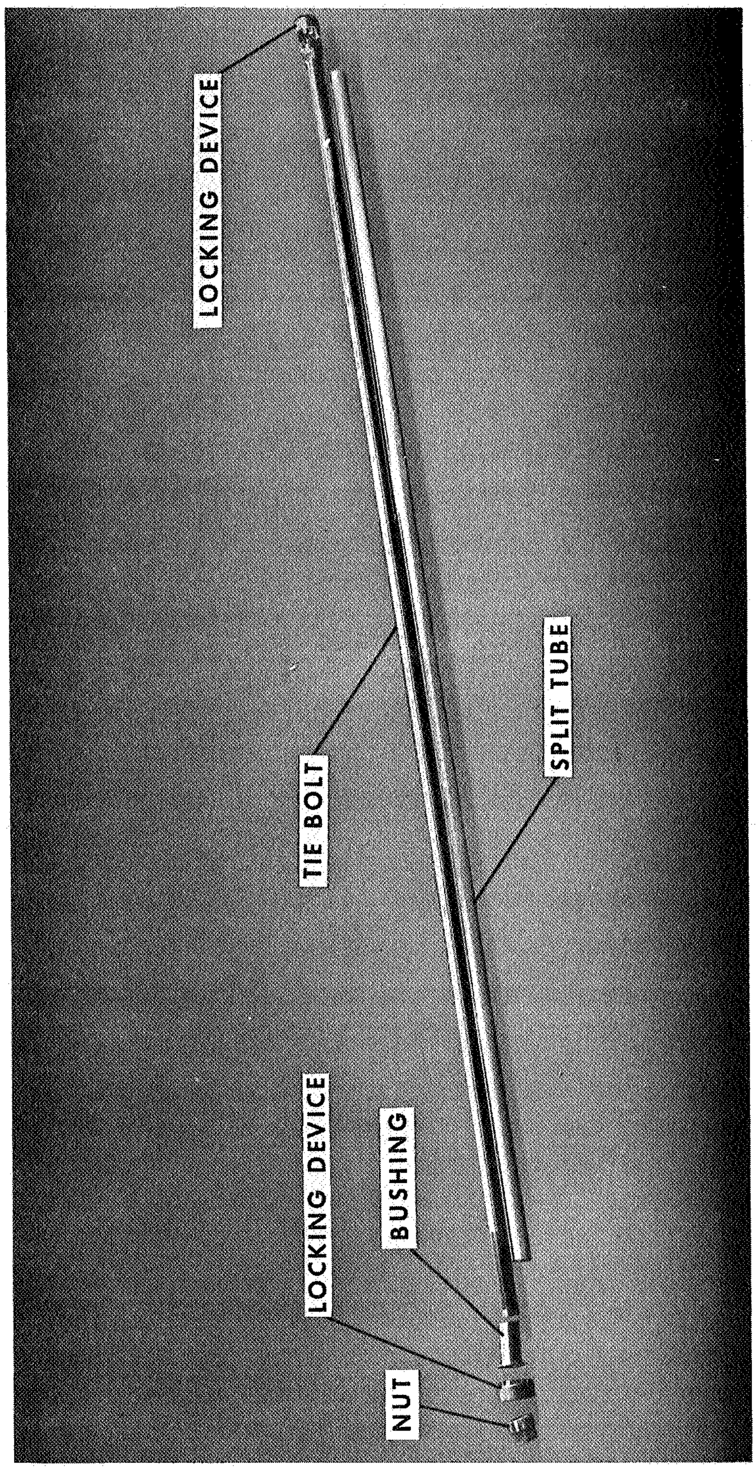

FIGURE 6-8. TIE BOLT COMPONENTS

when it is cooler than the sectors. Because this difference in temperature is small and the modulus of elasticity of titanium is low, the additional thermal stress is small in comparison with the preload.

The tie bolt preload is 10,000, ±500 pounds, and the maximum corresponding stress is 54,000 psi. Since the temperature and coefficient of expansion of the titanium tie bolt and beryllium sector match very closely, this value increases to only 55,000 psi at full-power, steady-state operation. The allowable tensile stress, based on 60% of room temperature ultimate strength, is 66,000 psi.

Environment. A mixture of liquid and gaseous hydrogen flows through the center hole of the tie bolt at temperatures ranging from 162° R to 183° R and pressures up to 750 psi.

D. SHIM RODS

Design Philosophy. These rods are required in the outer reflector for nuclear shimming (Figure 6-9). The basic design is the same for all of these rods; however, the rods are made of different materials. Specifically, beryllium rods are removed and aluminum rods inserted to change the nuclear worth of the outer reflector.

The shim rod is designed so that it can be easily installed and removed in the assembled reflector without disturbing other parts. This is accomplished by retaining each rod by means of its own locking device which is fastened to the nozzle end support ring with a special nut and bolt (Figure 6-10). The locking device retains the rod axially, while still permitting it to move radially so that the thermal motion of the sectors with respect to the nozzle end support ring is not restricted. There are 14 shim rods in each sector, making a total of 168 rods in all. These rods are always located as close as possible to the location that their counterparts occupied in the KIWI sectors; this placement provides proven experience as to their effectiveness prior to testing the reactor.

Description. The shim rod is a straight rod which is approximately 52 inches long and 0.497 inch in diameter. Near the nozzle end of the rod, there are two slots 180 degrees apart. These slots are provided for the shim rod locking device. On the nozzle end of each rod there is a slot resembling a screw driver slot; this opening is provided to allow coolant to pass through the shim rod bolt as it enters the plenum between the nozzle end support ring and the sectors. For identification purposes, all aluminum shim rods are anodized black. In addition to the anodize, all aluminum rods have an 0.060-inch diameter hole 1 inch deep in the nozzle end so that the rod can be identified even after it has been installed.

Material Selection. As with the sectors, beryllium is used for its neutron reflection capabilities. The aluminum rods, in turn, are used because they are essentially transparent to neutrons.

Design Analysis. Since the shim rod supports no other load than its own weight, its structural requirements are slight. The maximum load the slots on the nozzle end of the rod might experience is approximately 10 pounds. The stress developed in the rod or the locking device by this load is negligible.

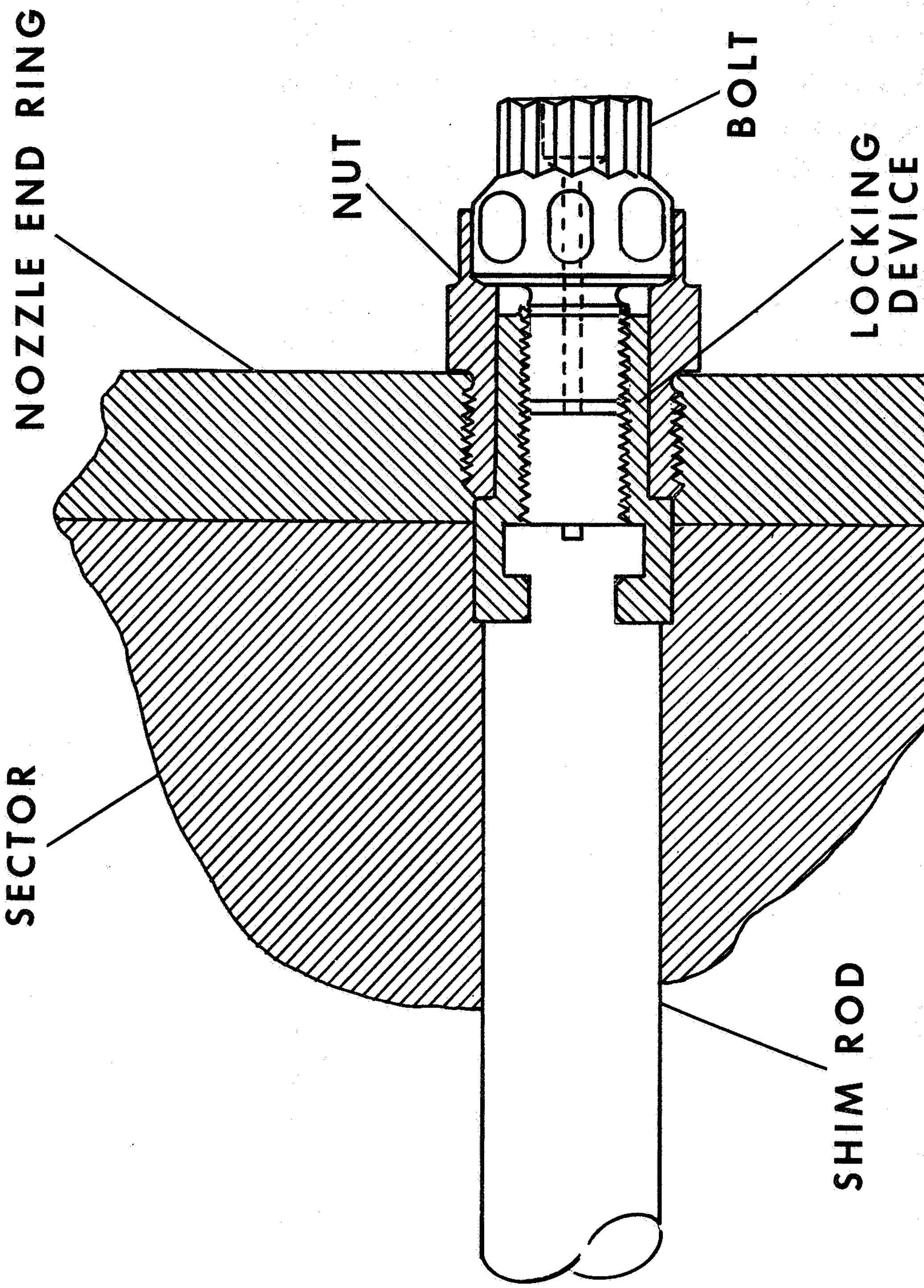

FIGURE 6-9. SHIM ROD

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Atomic Energy Act - 1954

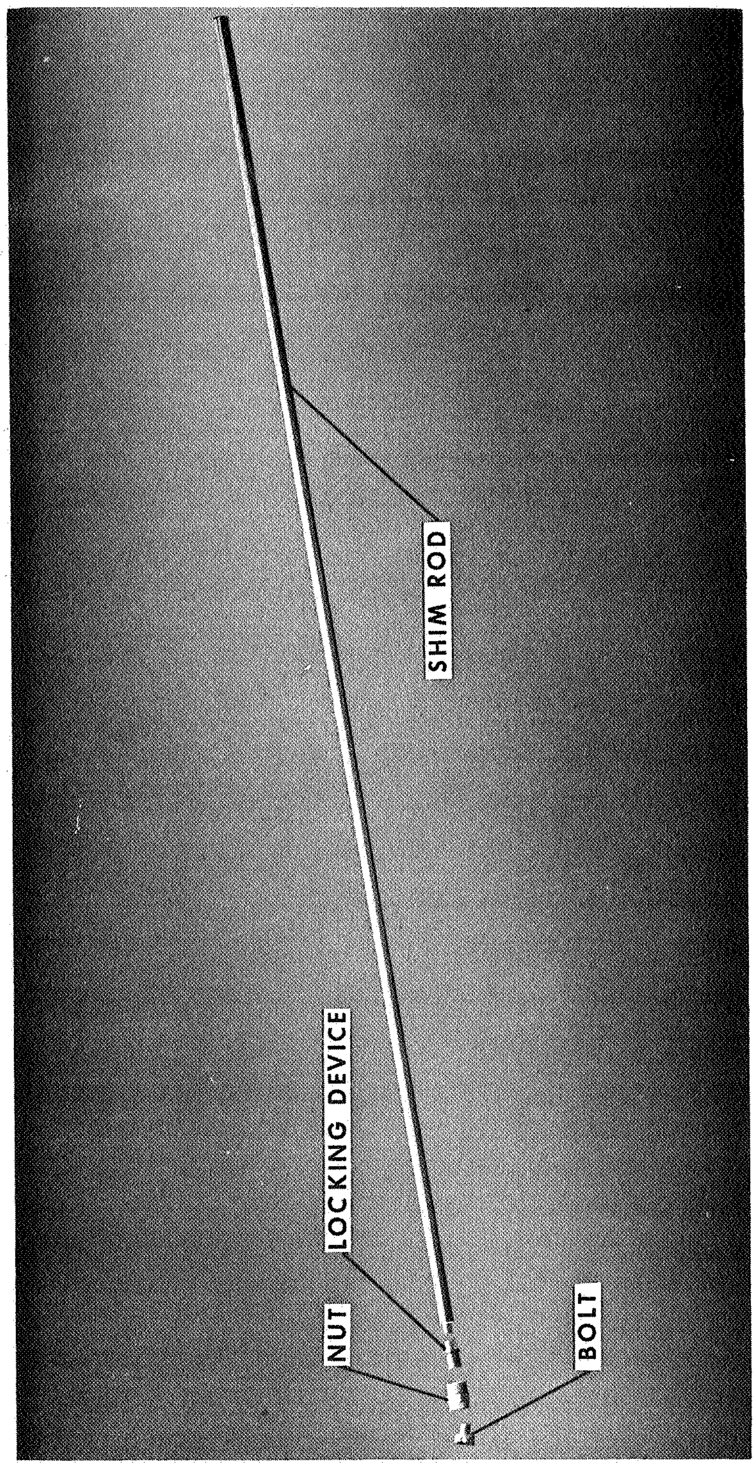

FIGURE 6-10. SHIM ROD COMPONENTS

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Atomic Energy Act - 1954

<u>Environment.</u> Because the shim rods fit tightly in the sectors, the coolant flow does not appreciably affect the shim rods. There will be some leakage through the clearance between the shim rods and the sectors, but the rods will be primarily cooled by conduction to the sectors. The shim rod temperatures, which are a function of the sector temperatures, range from 162° R to 325° R.

E. CONTROL DRUM

The control drums, which are located in the outer reflector, function by rotating a strip of poison toward or away from the core (Figure 6-11). Each of the reflector's 12 sectors has a control drum assembly which fits into the 4.214-inch diameter hole in the sector; this drum is the same length as the sector itself. The assembled unit is mounted so that it can be rotated from 0 to 180 degrees in order to control the reactivity of the reactor by turning the control plates which are mounted in the drum assembly toward or away from the core.

The control drum used for the Block I reactors differs from that used in the KIWI-B reactors in that it is encased in an aluminum housing, preventing any small loose pieces from dislodging and jamming moving parts or blocking flow passages.

There is an 0.050-inch radial clearance between the O.D. of the drum and the I.D. of the sectors. The clearance can be reduced by 0.017 inch because of thermal bowing of the drum during reactor operation. Flow in this area is controlled by reducing the annulus to 0.013 inch for approximately 0.062 inch at the dome end of the drum.

The major components used in the control drums are the beryllium cylinder, the dome end nozzle end bearing shaft extensions, the aluminum housing, related bolts, locking devices and springs and the plate assembly. Figure 6-12 shows these components.

A drive shaft coupling and stop are attached to the dome end shaft of the control drum. Proper orientation of the stop and coupling is provided by an uncut tooth on the shaft and a space on the mating parts. The stop engages a projection on the bearing housing and is held against this stop by a safety spring, insuring shutdown condition unitl the actuators are engaged. A positive pin lock which maintains the drum in the shutdown position is provided for shipping.

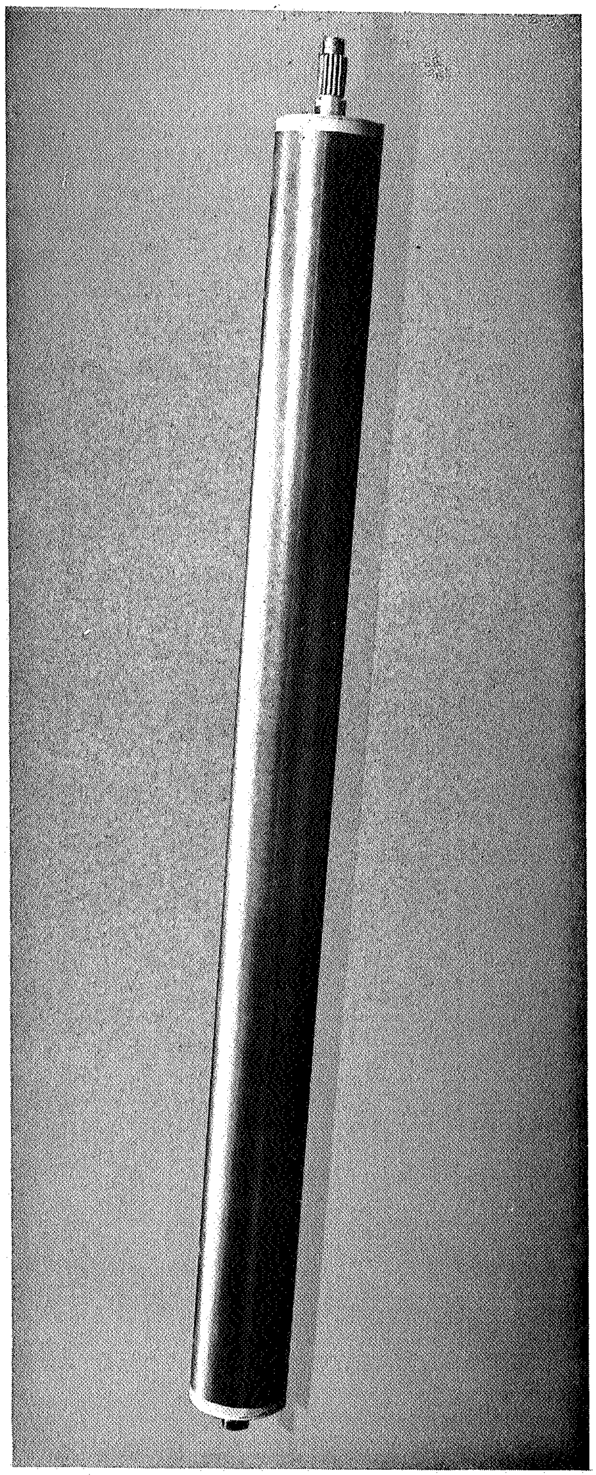

FIGURE 6-11. CONTROL DRUM

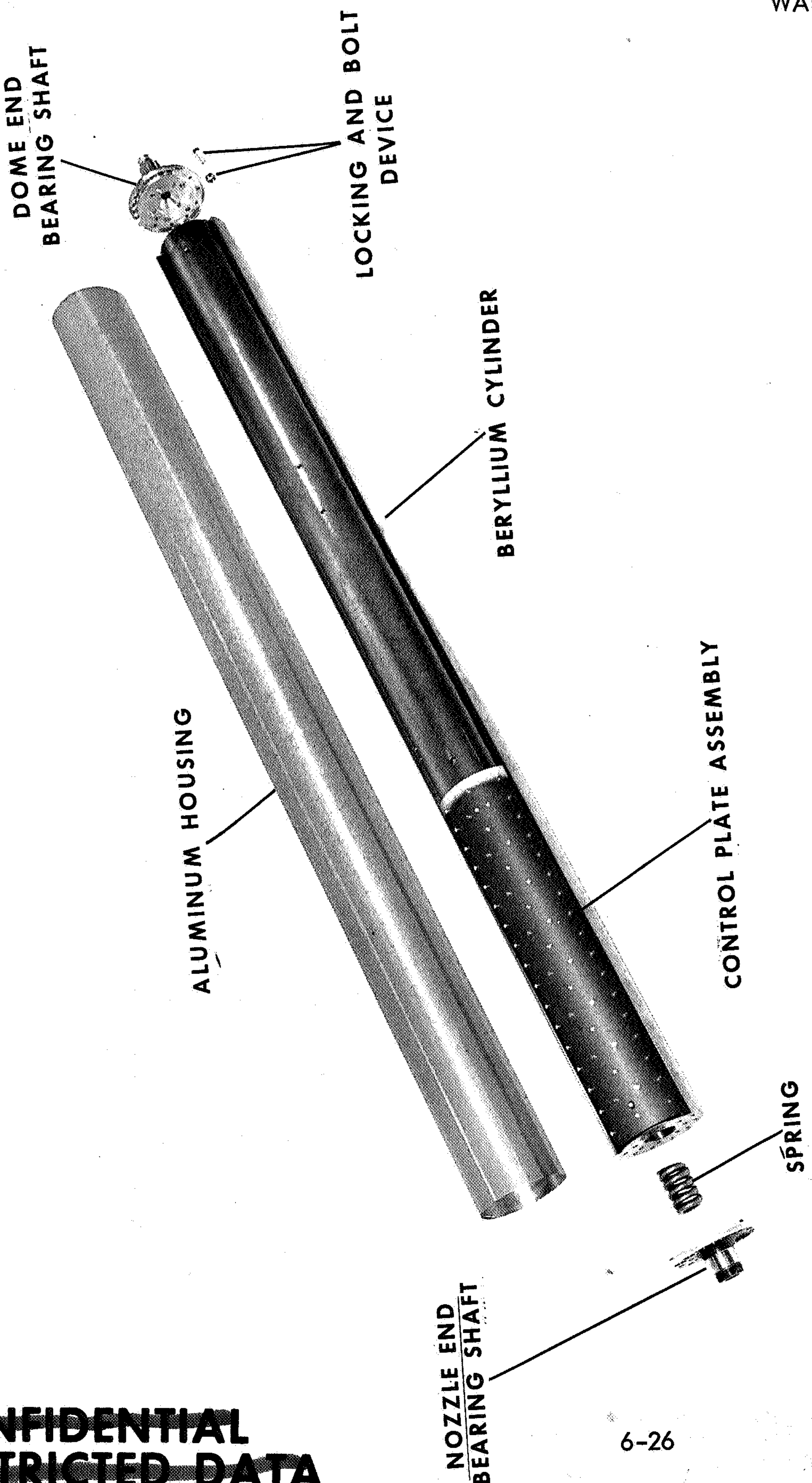

FIGURE 6-12. CONTROL DRUM COMPONENTS

1. Beryllium Cylinder

Design Philosophy. The control drum cylinder was designed to provide the maximum volume of beryllium in the control drums, while still allowing passages for sufficient coolant flow. In addition, an area is provided to accommodate the control plates.

To make certain that the control drum end caps are aligned properly with the flow passages and that the control plate is properly oriented with the actuators, one bolt hole in four is offset on the dome end. In addition, a hole is also provided at the nozzle end to accept a pin which is pressed into the nozzle end cap.

Description. The control drum cylinder, which is 52 inches long and 4.008 inches in diameter, has a relieved area 0.257 inch deep covering a radial area 126 degrees wide along the total length of the cylinder. Six holes are drilled in the center of the relieved area to form three sets. Each set has a 14.499-inch space between the holes. When the control drum is being assembled, pins are placed in these holes and the control plates are located and supported by them. Nineteen coolant holes are bored through the length of the cylinder. The aft end of the cylinder has a 1.380-inch diameter hole which is 1.290 inches in depth. The aft end also has a 0.064-inch diameter locating pin hole which is 0.353 inches deep. The pin is located 45 degrees off the vertical central line. The top of the cylinder has a 1.500-inch diameter by 0.060-inch high locating spigot, which pilots into the dome and shaft. Four Whitworth 0.250-20 B.S. threads are tapped 0.68 inch deep, with a 0.254-inch diameter by 0.187-inch deep counterbore in the top of the cylinder.

Material Selection. The control cylinder is made from hot pressed beryllium block. The selection of this material was made primarily because of nuclear considerations. Like the sectors, the selection of the material was based on LASL experience.

Dexign Analysis. In determining the most desirable position for coolant passages careful analyses were made to minimize thermal stresses in the cylinder. Whitworth threads are used in the tapped bolt holes to minimize thread stress concentrations. The tapped holes are counterbored approximately 0.1875 inch before the first thread. This arrangement provides a larger pull-out area for the bolt in tension.

Because of the pre-load torque, the axial load and corresponding tensile stress in the 0.025-inch stainless steel bolts that fasten the beryllium cylinder to the dome-end bearing shaft are 500 pounds and 28,000 psi. If the coefficient of friction between the aluminum bearing shaft and beryllium cylinder is as low as 0.1, the four bolts are required to transmit a small part of the 520 in.-lb. torque. This causes a peak shear stress in the bolt of 2000 psi, plus a bending and axial stress of 16,600 psi. The allowable peak shear and tensile stresses based on yield are 57,000 psi and 100,000 psi, respectively. Tests conducted at 160° R show that a load in excess of 6300 pounds is required to pull the bolts out of the beryllium cylinder. While bending moment and shear force transmitted by the bolt could reduce this margin, the reduction would not be significant. The thermal stresses around the cooling holes in the beryllium cylinder are less than in the beryllium reflector sectors and are, therefore, acceptable.

Based on the maximum calculated full-power, steady-state temperature gradient across the drum at any location, and assuming this gradient occurs over the whole length of the cylinder, the control drum bows 0.026 inch. The minimum radial clearance between the control drum and hole in the outer reflector sector, assuming the sector does not bow, is 0.043 inch.

Environment. A mixture of liquid and gaseous hydrogen at temperatures ranging from 162° R to 235° R and pressures up to 750 psi flows through the beryllium cylinder.

2. Dome End and Nozzle End Bearing Shafts

Design Philosophy. The control drum dome and nozzle end bearing shafts form the end closures for the control drum (Figure 6-12). These shafts have machined bearing surfaces which are used to center the drum within the sector. Both bearing shafts have through holes and plenum areas which regulate and locate coolant flow through the control drum; the larger holes are oriented so that the greater volume of coolant passes through the segment of the drum which contains the control plates.

The nozzle end shaft has a threaded portion at the end which retains the bearing. The end of the shaft that is toward the cylinder has a shallow hole which locates the control drum spring. The dome end shaft, in turn, has a threaded portion and a spline which is used to index and drive the control drum. The thread is used to attach the mating hardware (coupling, stop, bearing and bearing housing) to the control drum in the control drum assembly. Four through holes are provided in the dome end shaft to receive the bolts which have locking devices to hold the beryllium cylinder. The cylinder is also located by means of a hub on the cylinder and a shallow hole in the dome end shaft.

Description. The nozzle end bearing shaft is a plate 4.118 inches in diameter and 0.370 inch thick with a shaft extending 1.293 inches out on the side. The shaft has a 1.1815-inch diameter by 0.610-inch long bearing surface extending approximately 0.25 inch from the plate. The extreme end of the shaft is threaded with a 1.1730-inch 18HS-2A thread, and there is an 0.093-inch wide by 0.073-inch deep thread relief between the bearing surface and the thread. Twenty-one holes are provided in the plate to allow coolant to pass through the control drum. Fifteen of the holes are 0.185 inch in diameter; these holes are equally spaced over the plate. Five other holes, which are 0.500 inch in diameter, are located in a 120-degree radial area on the periphery of the plate. The remaining hole, which is also 0.500 inch in diameter, passes through the center of the plate and the shaft. The side of the plate opposite the shaft has a 0.125-inch deep plenum area through which all the coolant holes, except the 0.500-inch diameter center hole, pass.

The dome end bearing shaft is similar to the nozzle end bearing shaft except that it is splined to accept the drive shaft coupling for torque transmission to the drum. The spline surfaces are "hard coated" to prevent fretting and galling. Like the nozzle end shaft, it has several penetrations through the flange. These holes are used to meter the flow through the various parts of the drum. Separate plenums milled in the face of the flange which bolts against the beryllium cylinder allow for independent orificing of the various flow passages. A small lip on the outer diameter of the flange meters the flow in the annulus between the control drum and the sector.

Material Selection. The dome end and nozzle end bearing shafts are made from 6061-T6 aluminum because of its superior strength, thermal conductivity, weldability and neutronic transparency characteristics.

Design Analysis. The nozzle end bearing shaft is unloaded, except for radial bearing loads which are negligible. There are, on the other hand, appreciable stresses in the dome end shaft. The tensile stress in the thin section of this shaft, which is induced by the torque on the nut, is 4200 psi. The shear stress due to the 520 in.-lb. torque is 4900 psi. In combination, these stresses give a total stress intensity of 10,600 psi. However, since this stress is less than 25,200 psi, which is 60% of the ultimate strength of the 6061-T6 aluminum, it is acceptable. The spline stresses are the same as those in the coupling, and since the yield of the aluminum is higher (35,000 psi) they are also acceptable.

Environment. During steady-state conditions, a mixture of liquid and gaseous hydrogen at 162° R and 750 psia flows through the nozzle end bearing shaft. The dome end shaft operates in a mixture of liquid and gaseous hydrogen at 221° R and 733 psi during that period.

3. Aluminum Housing

Design Philosophy. The control drum housing is designed to transmit the torque loads in the control drum and to retain the control plates radially in the control cylinder recess (Figure 6-12). The housing also forms one wall of a partial annulus on the outside of the control plate for coolant flow passage.

This housing surrounds the beryllium cylinder and traps the control plate in its recess in the cylinder. A dome end shaft is welded to the housing and the beryllium cylinder is bolted to this end shaft. The relative position of the dome end cap and beryllium cylinder is insured by an offset retaining bolt. The nozzle end shaft is positioned relative to the beryllium cylinder by a locating pin; it is also welded to the housing. Bearing journals are machined after assembly of the housing, beryllium cylinder and end caps.

Description. The control housing is a cylinder 52.602 inches long with an O.D. of 4.118 inches and an I.D. of 4.014 inches. The housing is electron-beam welded to both the dome end and nozzle end caps.

Material Selection. The material used for the housing is 6060-T6 aluminum which was selected because it is essentially transparent to neutrons. The 6061-T6 grade was selected for welding compatibility with the dome end and nozzle end bearing shafts.

Design Analysis. The torsional stress in the housing caused by transmission of the drum torque is very low. The aluminum housing will shrink down on the control drum during operation. Under adverse tolerance build-up of the control plate and control plate-rivets, the housing may experience some local dimpling.

Torsional stress in the housing caused by the 520 in.-lb. torque is equal to 400 psi, and the critical torsional buckling stress is 9300 psi.

Since the aluminum housing has a larger coefficient of thermal expansion than the beryllium cylinder (Figure 6-13), the maximum diametrical shrink fit is 0.0030 inch. The maximum shrink fit occurs at housing and cylinder temperatures of 180°R. The resulting membrane stress is 7950 psi. The allowable membrane stress, based on 60% of the ultimate strength at 180° R is 30,000 psi.

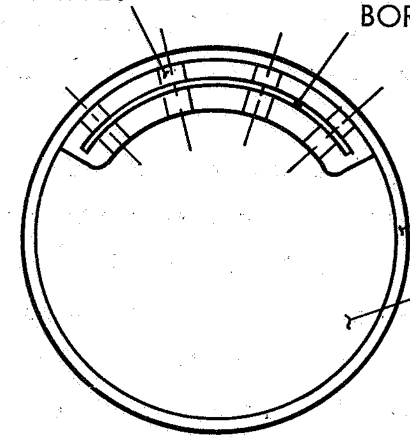

Figure 6-13. Typical Cross Section of the Control Drum

Environment. Liquid and gaseous hydrogen at temperatures ranging from 162° R to 231° R flows over the control drum housing at steady-steady operating conditions.

4. Spring

Design Philosophy. The control springs were designed to act as a back-up method of supporting the control drum cylinder should the cylinder or the bolts which normally hold it in place fail (Figure 6-12). This back-up design is required because of the possibility that the control plates, which are attached to the cylinder, could be radially misaligned if the cylinder fractured or the bolts failed in the control drum.

Description. The control drum spring is a 6-coil, cold-wound, helical spring with the ends squared and ground. The spring is wound from an 0.281-inch diameter wire. It has an O.D. of 1.281 inch (0.010 inch) and a free length of 1.895 inches.

Material Selection. Inconel X was selected as the spring material because of its high strength and good ductility. High strength was important because the space available for the spring was limited. This spring material has also proven to be satisfactory in the reactor environment.

Design Analysis. At control drum assembly, the spring is preloaded to approximately 160 pounds. This is equivalent to a 4-g load, which is the load the spring would experience if the control drum retaining bolts failed or the control drum parted radially. Since the spring is preloaded to meet this condition, it will maintain the drum in the desired position within the housing should the drum retaining bolts fail.

Under operating conditions, the helical spring experiences assembly preload deflection plus the additional deflection caused by differential contraction of the aluminum housing over the beryllium cylinder. The total deflection for the worst tolerance condition and an average temperature of 200° R in the cylinder and housing is 0.208 inch. These conditions would result in a torsional shear stress of 78,600 psi which is above the 60,000 psi shear allowable for Inconel X at the spring operating temperature of 160° R. Consequently, the spring may yield if all of the tolerances go in the wrong direction. However, beacuse of its its good ductility (20% elongation at 160° R) the spring will not fail. In addition, since the spring functions only as a back-up device for the bolts at the top end of the cylinder, yielding will not affect control drum operation.

Environment. The spring will operate in a liquid hydrogen environment at approximately 162° R to 180° R at reactor steady state operating conditions.

5. Plate Assembly

Design Philosophy. The control plates (Figure 6-12) are critical items because the nature of the material selected makes fabrication of a one-piece plate very difficult. Therefore, three identical plates are used in each control drum. Three plates are used to minimize the distortion that could occur because of axial thermal gradients. The plates have a slot at one end and a hole at the other. The hole and the slots are used to engage pins in the control drum cylinder in order to retain the plates. The slot permits the necessary movement of the plate relative to the control drum, this movement occurs because of temperature differences during reactor operation. The holes and slots are positioned to preclude improper assembly of the plates in the control drum. Rivets are located in a regular array on the plate surface. The rivets project from either side of the plate so that the plate can be positioned radially in the control drum recess to maintain the inner and outer annulus areas for coolant flow.

Description. The control plate, which is 17.276 inches long is formed to a 1.875-inch radius with an arc length of 3.978 inches. The plate material is 0.050 inch thick. Fifty-nine perforations in the plate accept spacer rivets. A hole at one end and a slot at the axial centerline are provided to engage the control plate retaining pins.

Material Selection. The control or absorber plate is made of 20% boron-10 (by weight) in a 99% pure aluminum matrix. Material selection was based on experience gained by LASL during the KIWI reactor tests.

Design Analysis. Under adverse tolerance build-up and temperature conditions, the control plate rivets can be loaded axially by shrinking the aluminum control drum housing against the rivet heads. Changes in length of the control plate caused by temperature change will be restrained by the rivets. This condition induces a maximum shear stress of 27,000 psi, which is well under the allowable shear stress of 29,000 psi.

Because of these loads, the maximum stress, including stress concentration around the rivet holes, is 7500 psi. Since the yield stress is 10,000 psi and only a few thermal cycles are expected, the boral plate is not expected to fail despite its extremely low ductility (measured values range from 0.5 to 10% at room temperature).

Environment. The control plate assembly is exposed to liquid and gaseous hydrogen at temperatures ranging from 162° R to 250° R and pressures up to 750 psi.

F. BEARING HOUSING

Design Philosophy. The bearing housing, which is shown assembled to the control drum in Figure 6-14, retains the control drum bearing at the outer relfector dome end. The housing has a concentric pilot or spigot which passes through the plate portion of the dome end support ring and into a counterbore in the sector. The fit is close (0.0005 inch) thereby providing axial and radial alignment of the control drum within the sector bore. An integral projection on the periphery of the bearing housing serves as a safety stop should the actuator stop fail. The projection also accommodates the safety spring sleeve and retaining bolt. Four ears on the housing are drilled and counterbored to accept the bearing housing retaining bolts and their Class A locking devices. Coolant passage holes are provided in the plate portion of the housing. The bearing housing as well as the major components of the drive train are shown in Figure 6-15.

Description. In order to accept the bearing outer race, the bearing housing bore is machined to a diameter of 2.4408 inches, ±0.0003. This tolerance provides a tight fit of 0.0001 inch and a loose fit of 0.0005 inch between the housing bore and the bearing outer race. A 2.750-18 NS-3B threaded section above the bearing bore accommodates a standard bearing retaining nut and lock washer. The diameter of the plate portion of the housing is about 4.28 inches. Four retaining bolt ears extend approximately 0.8 inch beyond this diameter. Ten 0.5-inch diameter coolant flow holes are symmetrically located in the housing plate section. At assembly, the overall height of the housing above the dome end support plate is about 1.1 inches.

Material Selection. The bearing housing is machined from an A-110 AT titanium forging. Titanium was selected because its coefficient of expansion is essentially the same as that of the beryllium sectors. Another reason for selecting titanium is the fact that the bearing housings are attached to the titanium dome end support ring and will, as a consequence, experience roughly the same temperatures; making the thermal stresses negligilble. The 440-C material used for the bearing races has a slightly higher coefficient of expansion than the titanium bearing housing; therefore, under operating conditions, the housing will not stress the bearing race.

Design Analysis. In addition to its obvious function, the bearing housing acts as a stiffener for the dome end support ring-plate at the control drum cutout. The stress due to this function is small. Although the exact temperatures in the bearing housing and the

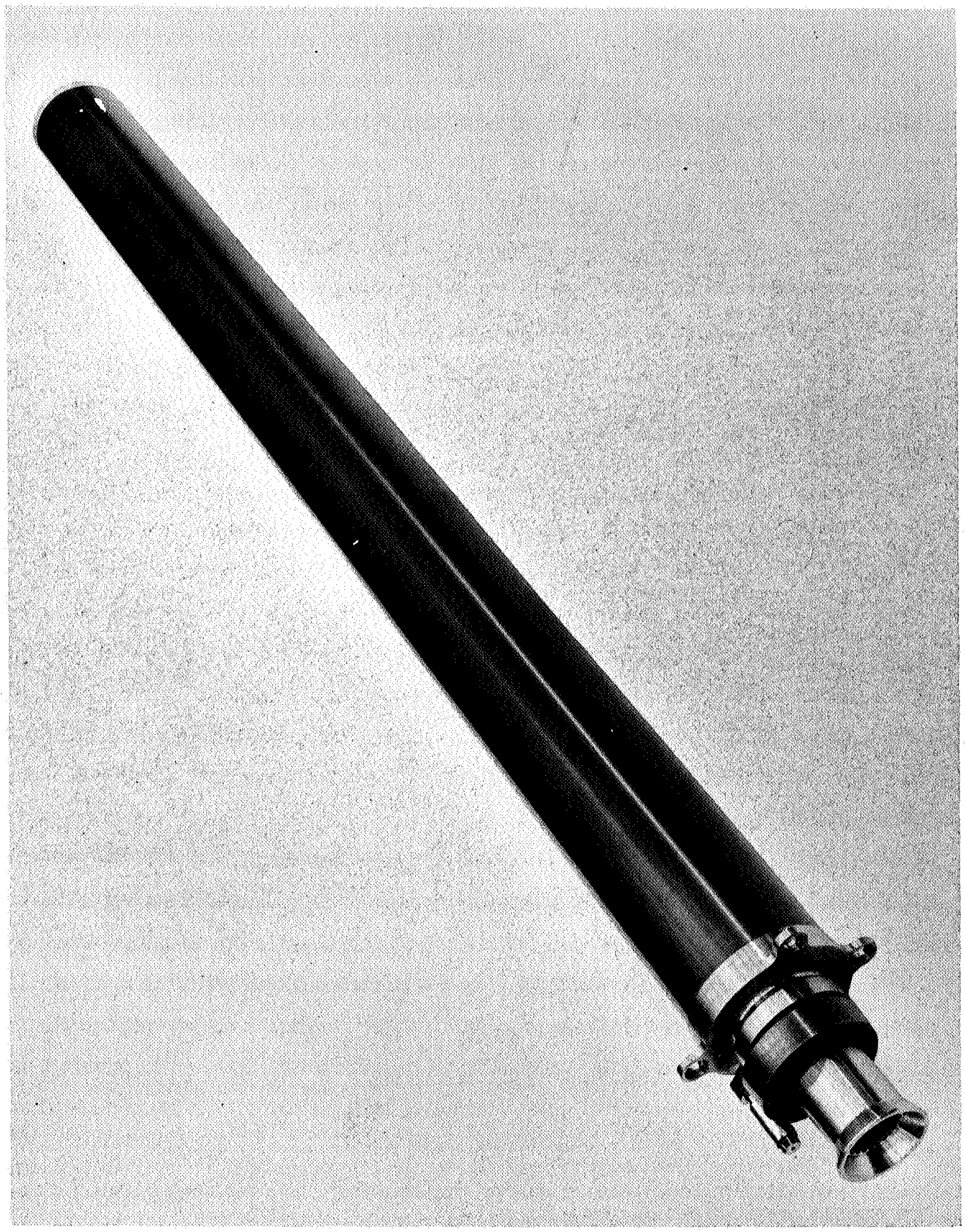

FIGURE 6-14. CONTROL DRUM DRIVE TRAIN COMPONENTS

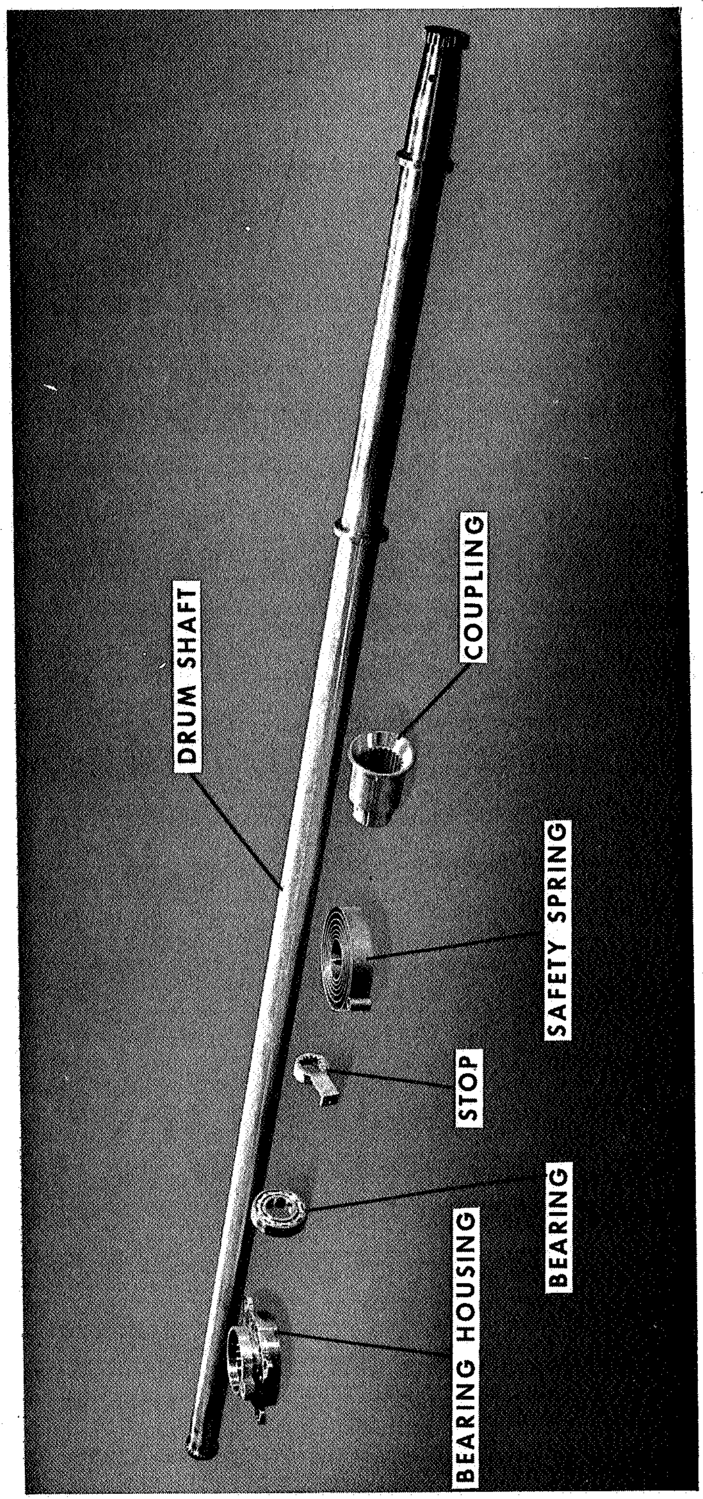

FIGURE 6-15. BEARING HOUSING COMPONENTS

adjacent plate have not been established, it is known that their difference will be less than 100° R. Accordingly, the thermal stress will be below 8000 psi (tension in the bearing housing and compression in the plate). This stress level is acceptable for the titanium housing.

Referring to Figure 6-16, it is seen that the peak shear stress at section A-A caused by torquing of the nut is 2400 psi. The bending stress at section B-B caused by axial pressure thrust load on the control drum is 37,300 psi, well below the 60% ultimate of 60,000 psi.

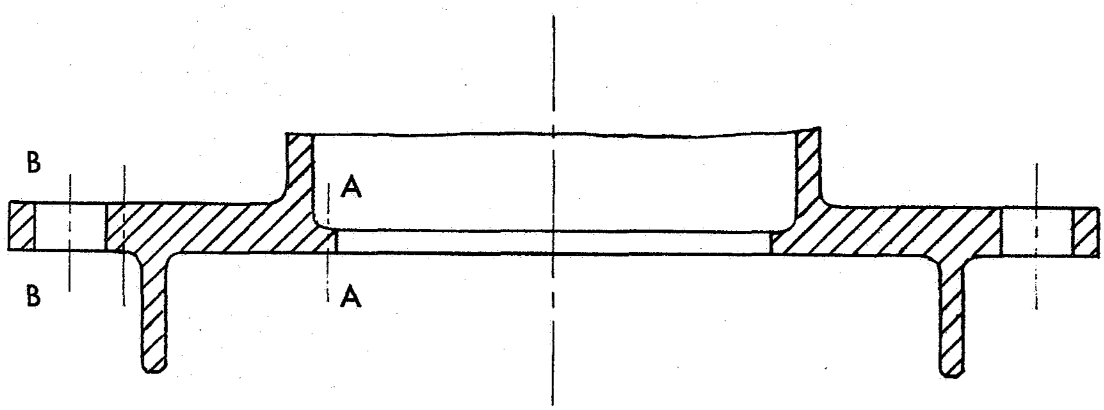

Figure 6-16. Control Drum Stresses

Environment. During steady-state conditions, liquid and gaseous hydrogen at 235° R and 700 psi flows through the bearing housing. During start-up, gaseous and liquid hydrogen at temperatures ranging from 162° R to 235° R could flow through the bearing housing.

1. Bearing

Design Philosophy. The control drum bearings for the NRX-A reactor must operate in a far more restrictive environment than conventional anti-friction bearings. They must withstand cryogenic temperatures, high levels of irradiation and high temperatures during decay heating. They must also maintain a fairly low uniform friction torque so that the actuators can perform adequately. All of these restrictive requirements must be met by a bearing which operates in an oscillatory fashion. Because of the multiplicity of requirements imposed upon the control bearings, a development program was initiated to develop a bearing which would meet these demands. Los Alamos Scientific Laboratory has developed ball bearings which were employed with apparent success in the KIWI-B reactor series. These bearings have withstood gaseous hydrogen and a fair amount of radiation from the B-1 reactor test. Based on LASL experience, the same bearing that was used in the KIWI reactors was choosen for the NRX-A. However the development program is continuing so that advanced bearing designs can be substituted as soon as they are evaluated and show demonstrable improvement.

Description. The bearings consists of eight 13/32-inch diameter balls housed between an inner and outer race. The balls are separated in the races by a two-piece, machined cage. The I.D. of the inner race is 1.811 inches and the O.D. of the outer race is 2.4409 inches. The axial thickness of the bearing is 0.6299 inch. Radial clearance between balls and races ranges from 0.0031 to 0.0037 inch. The cage has cutouts for gas flow.

Material Selection. The bearing inner and outer races and the balls are all made of modified 440C stainless steel (14% Cr and 4% Mo). The cage is AMS 4616 bronze, silver plated to a thickness of 0.0005 to 0.0015 inch. This material selection was based on LASL's experience with the KIWI reactors.

Design Analysis. The bearing design selected was based on bearings of various combinations of materials run in cryogenic environments. While the bearing loads during reactor operation are low, the bearings will experience shock and vibration during ground handling, shipping and boost operation. The bearing design, based mainly on low friction requirements, can withstand this shock and vibratory loading. The bearings will be subjected to a proof test.

Environment. The bearing operates in an environment which is a mixture of gaseous and liquid hydrogen with high neutron and gamma radiations. At steady-state conditions, the hydrogen flowing through the nozzle end bearing will be at 162° R and 750 psia. For the same conditions the hydrogen flowing through the dome end bearing will be at 235° R and 700 psia.

2. Coupling

Design Philosophy. The control drum coupling is designed both to transmit the driving forces from the drive shaft and to index the actuator to the control drum (Figure 6-15). The coupling uses two involute splines to transmit torque. The top of the coupling has a 30-degree coned flange to allow for blind assembly of the drive shaft into the coupling. The O.D. of the bottom portion of the coupling has a longitudinal slot which is used to position the control drum assembly spring.

Description. The coupling, which is 3.315 inches long, consists of two concentric cylinders positioned end-to-end; one 1.500 inches in diameter and 1.065 inches long, the other 2.060 inches in diameter and 1.750 inches long. A 30-degree angle flange, 0.500 inch long, extends out from the larger diameter. Both cylinders have involute splines on the I.D. Each spline has one tooth removed to provide for indexing. The maximum alignment between these teeth is 0° 10'. The larger spline is based on 26 teeth with a 1.625-inch pitch diameter, the smaller is based on 16 teeth with a 1.000-inch pitch diameter. The solid area between these two cylinders is bored out to a 0.790-inch diameter which is concentric to the splines. On the O.D. of the 1.500-inch diameter, there is a slot which is 0.044 inch wide, 0.140 inch deep, and 1.000 inch long extending up from the bottom of the cylinder.

Material Selection. The coupling is made of Type 304 stainless steel. The spline teeth are nitrided to provide increased wear resistance. This material was selected because it has adequate strength and is compatible with the 304 drive shaft and the aluminum control drum shafts.

Design Analysis. The coupling is conservatively designed. Maximum stress in the spline teeth when transmitting maximum drive torque is 5500 psi bending and 6900 psi in shear.

The maximum stresses of the two splines due to the 520 in.-lb. torque are illustrated in Figure 6-17.

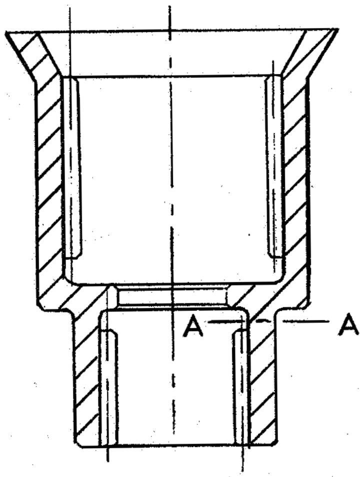

Figure 6-17. Coupling

Bending stress	=	5,300 psi
Allowable	=	30,000 psi
Shear stress	=	1,500 psi
Allowable	=	17,300 psi
Hertz stress	=	67,000 psi
Allowable	=	300,000 psi

Since these stresses are lower than the allowable, the coupling spline is adequate as designed. The shear stress at Section AA is 1180 psi.

Environment. At steady-state reactor operation, the coupling will be subjected to a mixture of liquid and gaseous hydrogen at a temperature of about 235° R.

3. Stop

Design Philosophy. The control drum assembly stop was designed to provide a positive mechanical stop for the control drum at 6 degrees beyond 0 degrees, which is the shutdown position (Figure 6-15). This function is performed by indexing the stop on the control drums with a keyed spline and providing a positive stop on the bearing housing which is bolted to the dome end support ring. The stop is also used as part of the positive locking device which locks each control drum in a shut-down position when the reactor is assembled without the actuator installed. This locking is accomplished by passing a pin through the stop and into the bearing housing, where it is secured by a crimped class "A" locking device.

Description. The control drum assembly stop has a cylindrical portion which is 1.5 inches in diameter and 0.600 inch long. An L-shaped arm, 0.750 inch wide and 0.170 inch thick, extends out from the top side of the cylinder to a point 1.970 inches from the center of the cylindrical section. The base of the "L" is at the farthest point from the cylindei (pointed down); it is 0.600 inch long, 0.188 inch wide, and has a 0.125-inch diameter hole centered on the width, 0.410 inch from the top of the part. The I.D. of the cylinder is an involute spline with a 1.00-inch pitch diameter. The spline is based on a 16-teeth configuration with one tooth removed to form an indexing slot.

Material Selection. Cold worked Type 304 stainless steel, with a minimum yield of 100,000 psi was selected for the stop. This cold worked material was selected because of its high strength and its thermal compatibility with the aluminum control drum shaft.

Design Analysis. Primarily, the stop serves to maintain the control drums in the shut-down position while the reactor is going through shipping and the ground handling phases. Its secondary function is to stop the control drum in the shut-down position if the drive shaft fails or the actuator malfunctions (which would permit the safety spring to come into play).

Environment. The stop will be exposed to a mixture of liquid and gaseous hydrogen. Temperature of the gas around the stop at steady-state operating conditions will be about 235° R.

4. Safety Spring

Design Philosophy. The control drum springs were designed to keep the control drums in the shut-down position during the time when the locking pins have been removed and the actuators are being installed (Figure 6-15). The I.D. of the spring is held by a tang which slips into a slot in the control drum coupling. The loop on the O.D. is held by a sleeve and a bolt which orient the spring with respect to the bearing housing and keep its coils separated to facilitate cooling.

Description. The control drum spring is a torsion-wound spring having approximately 8 turns. The cross-section of the spring material forms a rectangle which is 0.875 inch long x 0.040 inch thick.

At the inside diameter, the end of the spring is bent over to form a tang which locates it with respect to the control drum. On the outside diameter the end of the spring is rolled into a loop which is bolted to the fixed bearing housing.

Material Selection. The material selected for the spring is AMS 5519 stainless steel. This is a cold worked 304 type material having a yield strength of 140,000 psi. The material was selected because it has been proven to be a satisfactory spring material for the reactor environment.

Design Analysis. The maximum stress in the AMS 5519 stainless steel bias spring is 108,000 psi at its maximum rotation of 180 degrees. The room temperature yield stress is 140,000 psi.

The bending stress in the bolt caused by the spring load is 8600 psi. The axial stress due to torquing the bolt is 18,500 psi. The 27,000 psi which results is below the room temperature yield of 30,000 psi for the 304, condition"A", stainless steel bolt (Figure 6-18).

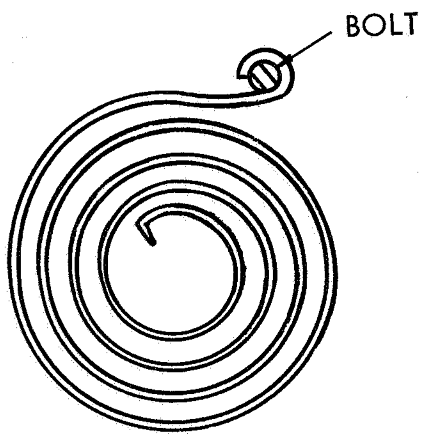

Figure 6-18. Safety Spring Bolt Design Analysis

Environment. The spring will be subjected to a mixture of liquid and gaseous hydrogen. Temperature of the gas around the spring at steady-state operating conditions will be approximately 240° R.

5. Drive Shaft

Design Philosophy. The drive shaft was designed to transmit driving forces from the actuators to the control drum via the control drum couplings (Figure 6-15). The shaft has a crowned tooth splined gear at each end, a design which allows free rotational movement of the shaft even if a misalignment as great as 2 inches over the length of the shaft should occur. The drive shaft and basic drive train components are shown in Figure 6-15.

Description. The drive shaft is 72.195 inches long with an O.D. of 1.600 inches and an I.D. of 1.100 inches. The O.D. tapers from 1.600 to 1.500 inches in 2.25 inches at the actuator end, and from 1.600 to 1.250 inches in 3.00 inches at the control drum assembly. The I.D. of the shaft tapers from 1.600 inches to 0.750 inch in 1.93 inches, then remains at 0.750 for 1.07 inches at the control drum end of the shaft. The shaft has two 0.250-inch holes through it to allow coolant to flow from the control drum into the plenum between the shield and the outer reflector. These holes, which are located 180 degrees apart, are at 2.00 inches and 1.500 inches from the control drum end. There is a crowned tooth spline at each end of the shaft. To allow for indexing, each spline has a tooth space which is not cut out. The maximum misalignment between these two spaces is 0° 10'. At the control drum end, the spline has a 1.6250-inch diameter spherical pitch and, based on a 26-tooth pattern, has 25 teeth. The spline at the actuator end of the shaft has a 1.750-inch spherical pitch diameter and 27 teeth, based on a 28-tooth pattern.

Material Selection. The drive shaft is made from Type 304 stainless steel. This material was selected because its mechanical properties are adequate and because of its proven performance in nuclear systems.

Design Analysis. The total inertia load of the control drum, coupling and extension shaft is 0.26 in.-lb./sec^2, and the maximum acceleration or deceleration of the drum is 2000 radians per/sec^2. Under these conditions, the maximum torque is 520 in.-lb. The resulting shear stress at the minimum cross section of the shaft is 1500 psi. Based on membrane stress, the allowable room temperature stress for this shaft is 13,900 psi.

The splines are nitrided to keep the surfaces from galling and to increase the allowable contact stresses. The spline end of the shaft that drives the control drum is the smaller of the two splines; its stresses are tabulated below:

Stress	Value (psi)
Bending	4,200
Allowable	30,000
Shear	1,400
Allowable	17,300
Hertz contact	48,000
Allowable	300,000

The shaft is adequate as presently designed, and the spline is satisfactory since it is not subject to repetitive-type loads.

Environment. The drive shaft operates in liquid and gaseous hydrogen at 241° R and 706 psi, during steady-state conditions.

G. STRESS ANALYSIS OF THE OVERALL OUTER REFLECTOR ASSEMBLY

In addition to the individual stress analysis discussed so far in this chapter, a compilation of full-power, steady-state stresses and deflections encountered during operational conditions for all the outer reflector components, except locking devices and bearings, was released in WANL-TME-496, NRX-A Outer Reflector Assembly Analysis. Detailed reports, including calculations on the individual components are now being systematized and a formal transient stress analysis will be started as soon as the transient temperature distributions are established.

Figure 6-19 is an isometric drawing of one sector which was analyzed as a representative section of the outer reflector. As a result of these analyses, it was concluded that the outer reflector assembly will function properly for steady-state operations.

Structurally, the outer reflector assembly is composed of 12 heavy beams (beryllium sectors) sandwiched between 2 titanium rings (dome and nozzle end support rings). The radial expansion and rotation of this assembly is primarily determined by the gross deflection of the two rings. Figure 6-19 presents a schematic cross-sectional representation of the outer reflector assembly indicating the nominal temperature and external mechanical loads encountered during full-power, steady-state operation.

Since calculations show that localized deflections are small, they have been neglected. The deflection at point "A" on Figure 6-19 in the radial "r" direction is listed in the table as "Δr_A"; rotation at point "A" is represented by "θ_A". All deflections in Table 6-1 are referenced to the condition where the outer reflector is at a uniform temperature of 70° F, has no tie bolt preload, has no externally applied mechanical load, and experiences no gravity load. The deflections and rotations in the last column of the table are the net deflections obtained by adding the values in the first three columns.

TABLE 6-1

Full-Power Steady-State Gross Deflections

	Tie Bolt Load	Operating Temperature	External Mechanical Full-Power Steady-State Loads	Net Values
Δr_A	nil	-0.0110 in.	-0.0063 in.	-0.0173 in.
Δz_A	nil	-0.0204 in.	+0.0019 in.	-0.0185 in.
θ_A	-0.000284 rad.	-0.000367 rad.	+0.000147 rad.	-0.000798 rad.
Δr_B	-0.0037 in.	-0.0535 in.	+0.0098 in.	-0.0333 in.
Δz_B	-0.0005 in.	-0.0535 in.	+0.0019 in.	-0.0521 in.
θ_B	nil	-0.000322 rad.	-0.0000023 rad.	-0.000299 rad.
Δr_C	nil	-0.0277 in.	+0.0056 in.	-0.0333 in.
Δz_C	-0.0009 in.	-0.0900 in.	+0.0019 in.	-0.0890 in.
θ_C	+0.000284 rad.	-0.000277 rad.	-0.000192 rad.	+0.000199 rad.

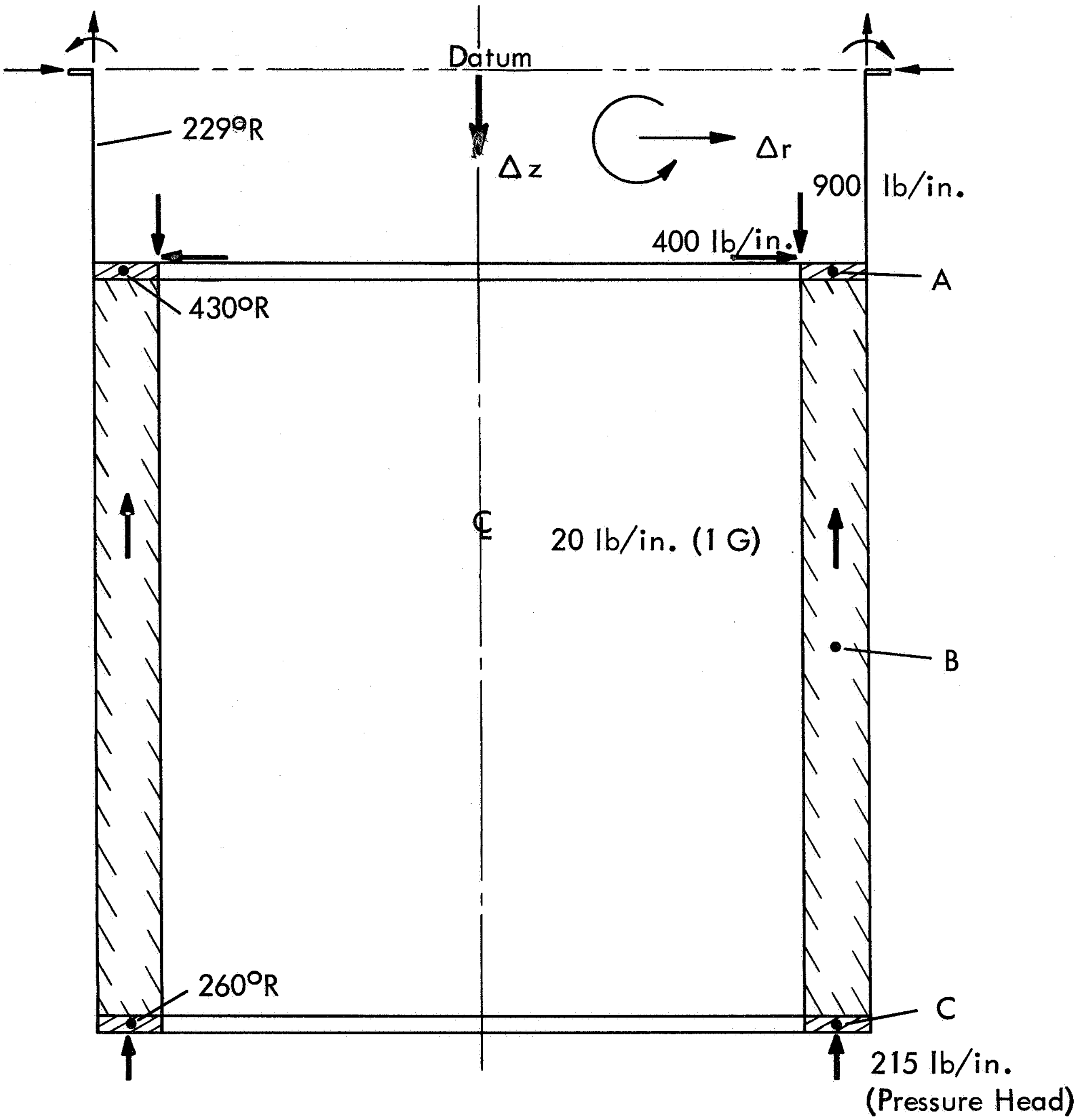

FIGURE 6-19. LOADS AND TEMPERATURES

VII. CORE ASSEMBLY

The core assembly is a unit which is approximately 3-1/2 feet in diameter and 5 feet in length (Figure 7-1). The assembly is composed of two closely related geometrical areas: the core subassembly and the inner reflector. The core assembly encompasses both of these areas in an assembly which performs the following functions:

(1) Provides the heat source and acts as a heat exchanger for the rocket propellant.

(2) Absorbs axial and lateral acceleration loads.

(3) Maintains the fuel elements in a bundled configuration.

(4) Distributes coolant flow for proper heat dissipation.

The core subassembly (Figure 7-2) is made up of 199 regular clusters each of which, in turn, is composed of 7 elements, plus 42 irregular clusters which convert the periphery to a more nearly cylindrical shape. These clusters are fastened to the support plate by four different fuel cluster nut configurations. In all, 289 of these hollow nuts are used. They are held in place by crimped locking cups which keep them from loosening under vibration loads. A series of 162 filler strips complete the transition of the core periphery to a cylindrical configuration. At the flow inlet end of the core, filler blocks replace the filler strips for approximately 4-1/2 inches. A titanium band provides a bundling pressure of about 10 psi at this point. The band also keeps the inter-element gaps from changing during shipping or operation. The filler blocks are keyed to the core elements, while the filler strips are free to slide axially relative to the core.

The core elements are positioned about 1-5/16 inches from the core support plate, but the tie rod, with an attachment point which is about 5 inches above the fuel, provides lateral flexibility of the core to compensate for shifts caused by lateral shock loading or thermal expansion. The dome end seal closes off the periphery of the plenum between the support plate and the fuel elements by bridging the gap between the support plate and the filler blocks. The flexible seal fingers also allow lateral shift of the core relative to the core support plate.

The core subassembly is both the heat source and heat exchanger for the rocket propellant. The coolant (propellant) flow passes through the many support plate holes into the plenum between the core and support plate and then through each cluster assembly, where it is heated prior to leaving the core. The pressure drop introduced across the core elements is about 130 psi.

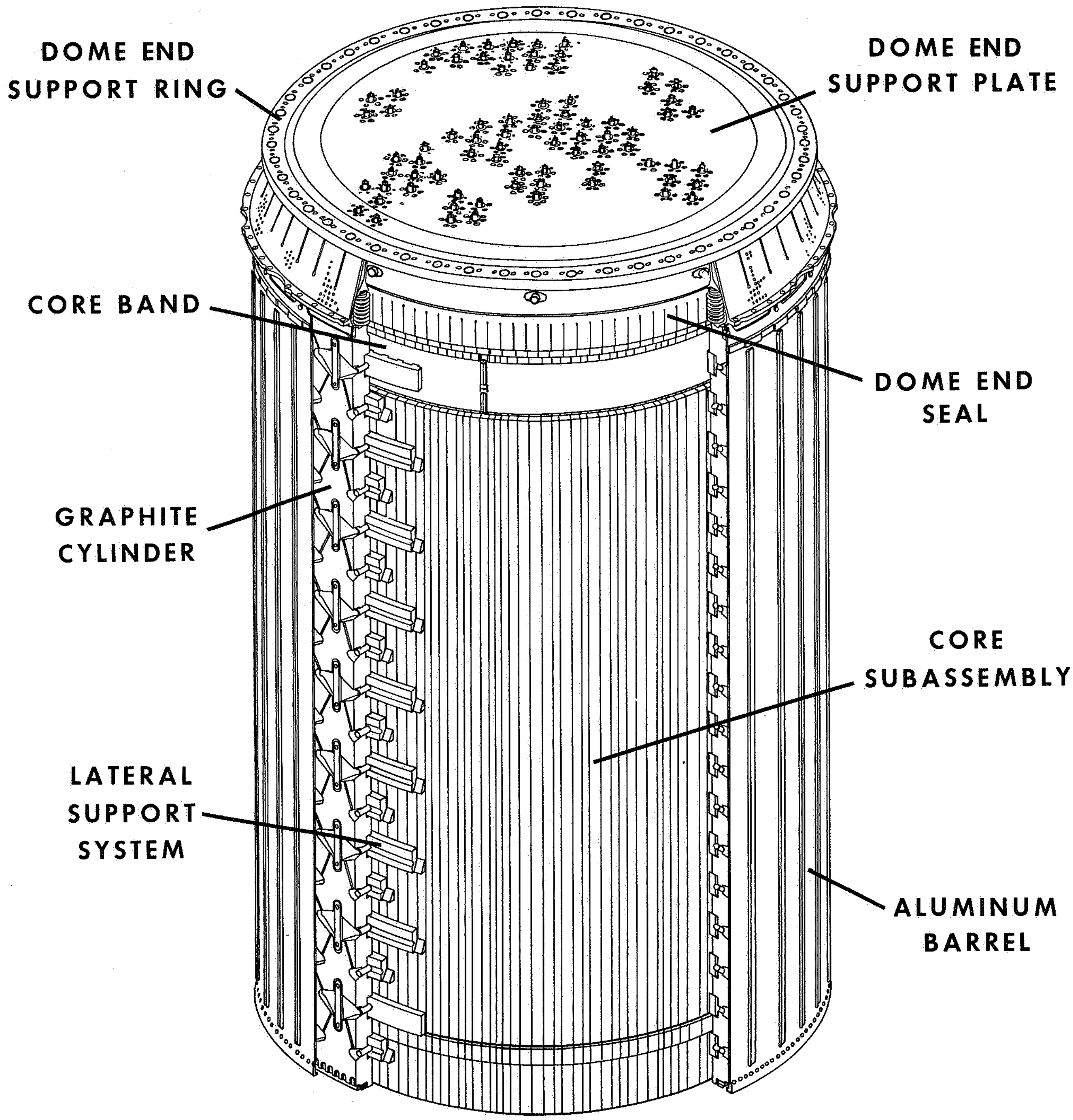

FIGURE 7-1. CORE ASSEMBLY

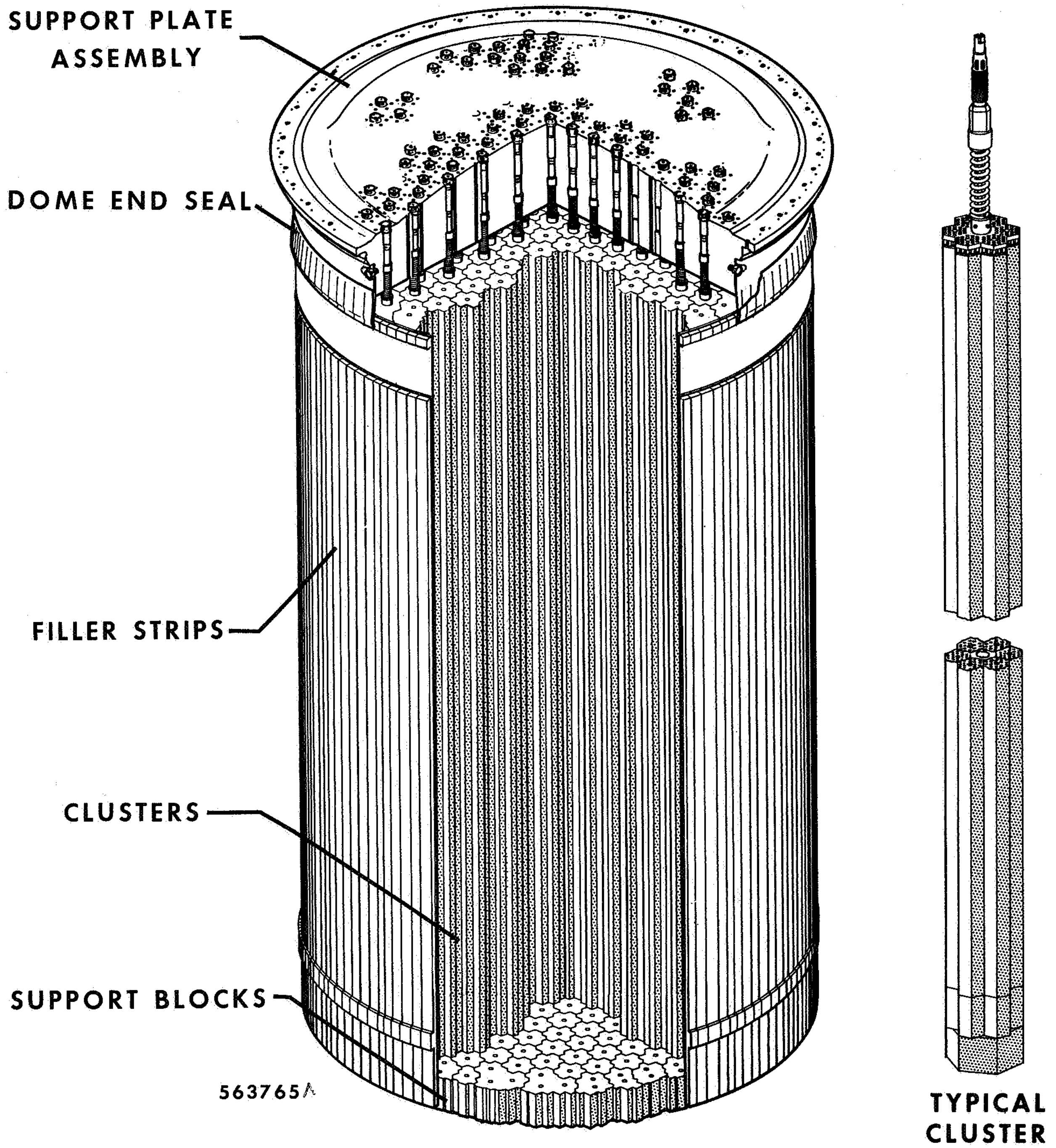

FIGURE 7-2. CORE SUBASSEMBLY

Figure 7-3 shows the portion of the core assembly which is most commonly called the inner reflector. While this unit, which is located between the filler strip of the core periphery and the outer reflector, fundamentally serves as a neutron reflector, it also serves the following additional functions: it provides a bundling force and a lateral support for the core; it acts as a support for the filler strips of the core subassembly; and it acts as a pressure barrier. The inner reflector's location between the core and outer reflector introduces many material restrictions on components, particularly from a neutronics stand-point. For example, large amounts of neutron absorption in this area would seriously affect the control of the reactor since the control drums are located in the outer reflector. The high temperatures experienced in the area adjacent to the core during operation and decay heat impose further restrictions on the material selected for the design.

Sealing is provided between the nozzle inlet and core exhaust by a bellows-type seal which is acted upon by a coil spring, loaded, graphite cylinder. Close tolerance fits on the plungers passing radially through the cylinder restricts leakage flow into the lower pressure areas adjacent to the core.

Lateral support and additional sealing are accomplished by a distributed set of segmented seals which are spring-loaded against the core by a series of plungers and leaf springs which are attached to the graphite cylinder. There are 18 rows of seal segments, of which two are inactive as seals. Axial slots on the seal segment inner diameter provide a controlled pressure distribution on the outside of the core. Lateral loads are transmitted from the seal segments, through an aluminum barrel and to the outer reflector.

The outward projections on the filler strips fit between the two rows of seal segments at the forward end of the cylinder and serve to keep the filler strips in the proper axial location against the core.

Axial core loads are transmitted to the outer reflector by a cone-shaped core support ring which is located between the core support plate and outer reflector dome end support ring.

An aluminum barrel surrounding the graphite cylinder meters flow in the annulus between the core assembly and outer reflector. This metering is done by means of a ring attached to the barrel. Spacers attached to the barrel guarantee that the flow area in the annulus is not reduced below a minimum value and that eccentricity of the core assembly with respect to the outer reflector is limited.

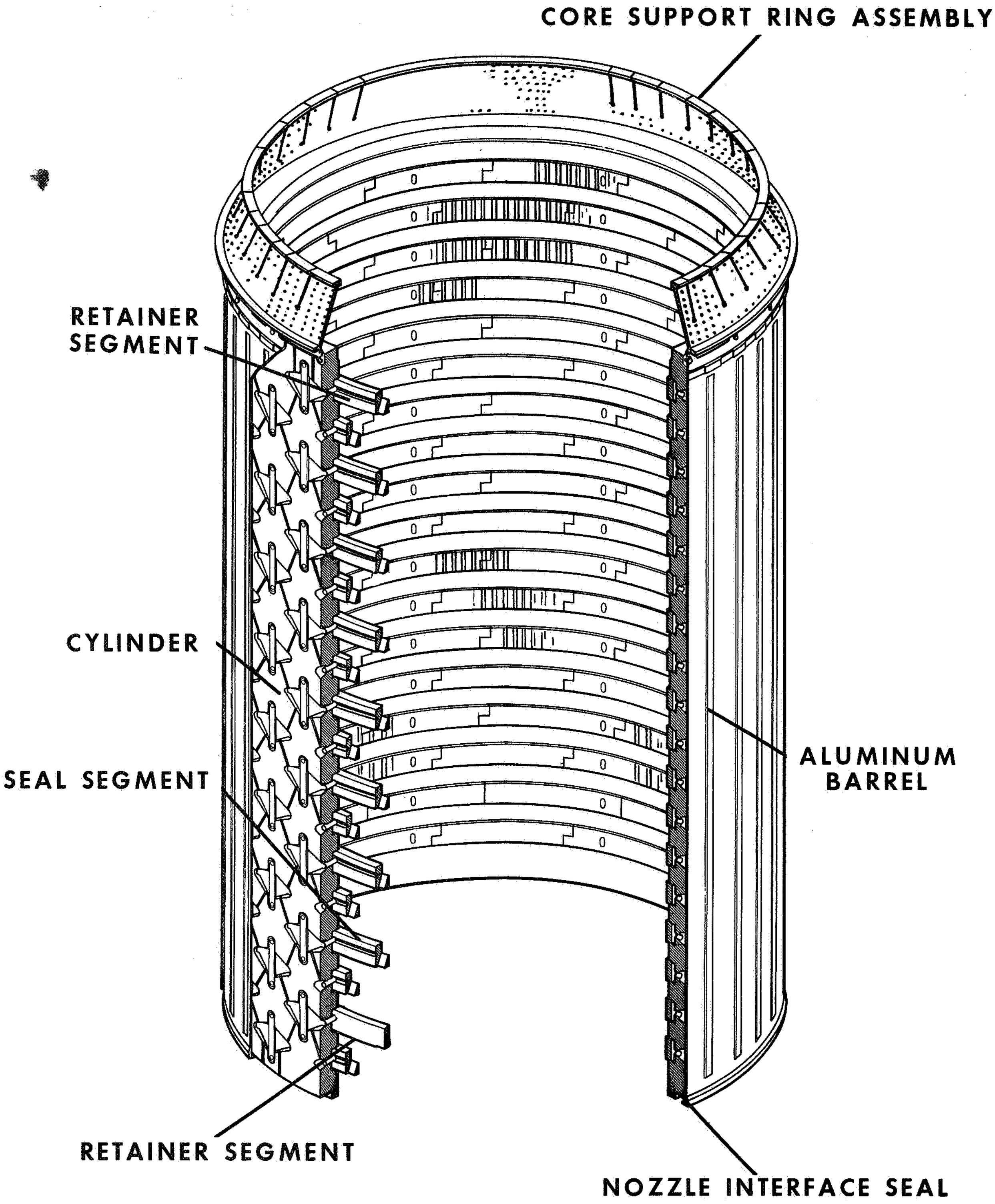

FIGURE 7-3. INNER REFLECTOR

The balance of this chapter discusses the specific subassemblies and hardware items which comprise the core assembly.

A. SUPPORT PLATE

Design Philosophy. The core support plate provides the attachment point for all of the fuel cluster assemblies. While both the positive and negative axial loads of the fuel element bundle are carried through the support plate, the lateral loads are reacted through the lateral support system. The flow inlet surface of the plate is curved both for compactness and optimization of stress distribution. The large number of coolant holes minimizes thermal stresses by minimizing temperature differences throughout the plate. Slots are broached into the plate at the fuel cluster attachment points to prevent rotation of the tie rod holder while the cluster nut is being tightened. Figure 7-4 shows the core support plate with the dome end seal in place.

Description. The core support plate is just over 3 feet in diameter, with a thickness which is contoured from 6 inches at the center to approximately 4 inches near the periphery. One face of the plate is flat, and the thickness variation affects only the flow inlet face. At the outer periphery, there is a 0.625-inch thick bolting flange which mates with the core support ring; 48 bolts join these two parts. The plate contains approximately 2500 holes which provide coolant passages and allow for insertion and attachment of the fuel clusters. The coolant holes, which constitute the major number of these, are 0.375 inch in diameter. A pin pressed into the peripheral flange provides angular orientation relative to the conical support ring.

Material Selection. This plate is machined from a contoured forging which is made of Type 2219 T56 aluminum. The material provides a higher temperature margin on yield strength during the pulse cooling period of reactor operation.

Design Analysis. The support plate, which is supported at its periphery, deflects just under 0.1 inch during reactor operation; about 3/4 of this deflection is due to thermal bowing. Peak local stresses near the center of the plate are under 15,000 psi, with the yield strength at about 44,000 psi.

Environment. During steady-state operation, the support plate temperature is approximately 560° R, with a maximum axial gradient of 130° R and a radial variation up to 560° R. During pulse cooling, the plate may reach 1050° R.

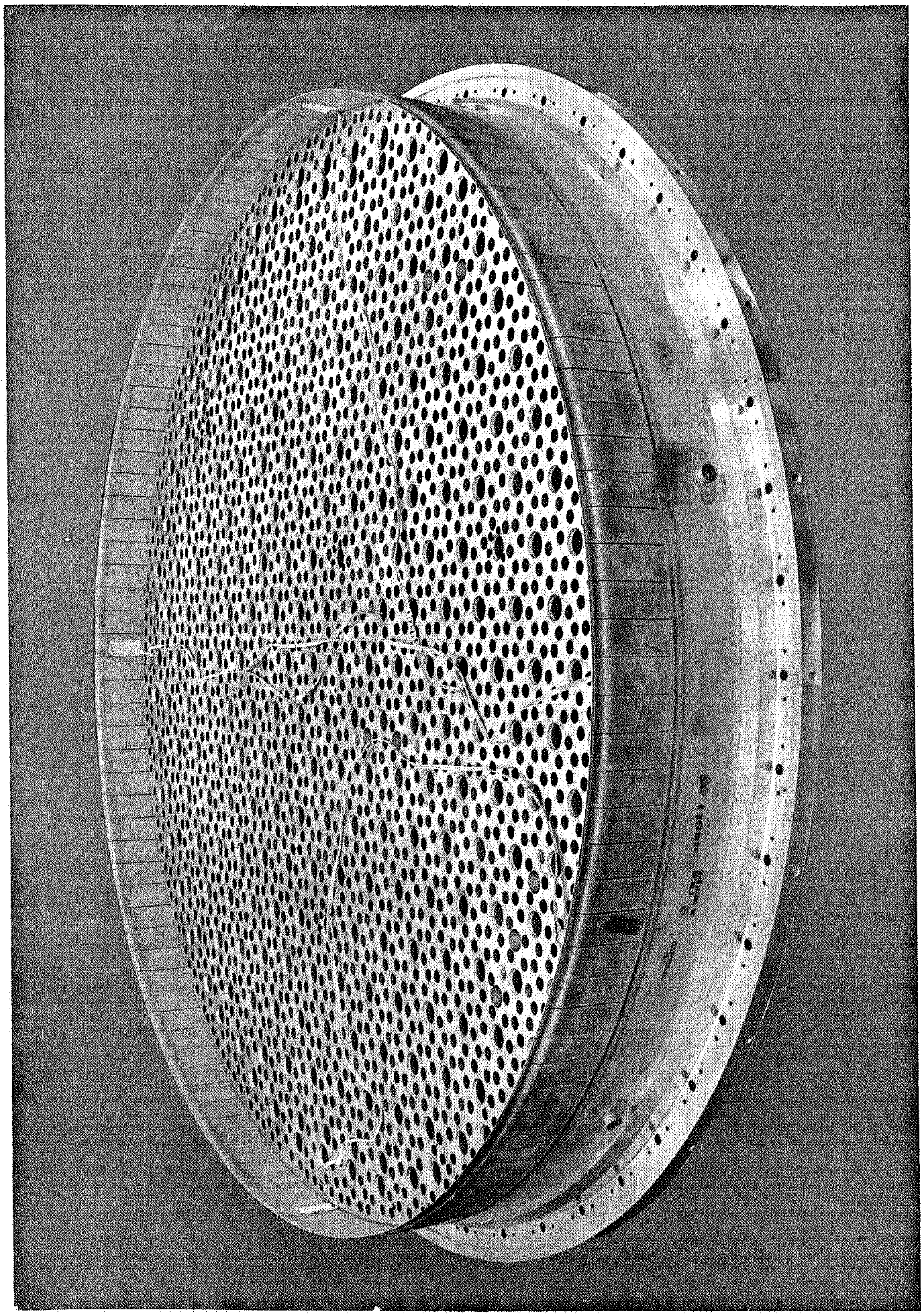

FIGURE 7-4. CORE SUPPORT PLATE

B. DOME END SEAL

Design Philosophy. This unit allows the core to move (relative to the core support plate) while still providing an adequate coolant seal at the point of contact with the perimeter of the core. The seal prevents coolant leakage from the reflector area into the core inlet plenum.

Description. The unit consists of two concentric cylindrical rings. The material is explosive-formed in dies from flat sheet which is approximately 0.032 inch thick. It is then fusion-welded into two rings. The two rings are fitted concentrically, one inside the other; and axial saw cuts, approximately 0.010 inch wide, are made around the circumference at points approximately 1 inch apart. The rings are then rotated relative to one another so that the saw cuts on the inner and outer rings are staggered 0.5 inch apart, with each gap sealed by the solid portion of the adjacent ring. When the rings are then welded together, the saw cuts form spring fingers which expand radially outward and act as a seal when installed over the bundled core.

The seal is press-fitted over the core support plate and firmly attached by means of six socket-head screws captured by Class A locking cups; this whole assembly comprises the core support plate assembly. This seal, which is shown in Figure 7-5, bridges the gap between the support plate and the core assembly.

Material Selection. The seal is fabricated from Type 2219-T31 aluminum sheet to minimize differences in thermal expansion between it and the core support plate. A new heat-treatment process was developed to combat a rather unique problem in fabrication. It was found that during the process of explosive-forming the parent material was cracking in the vicinity of the fusion weld. It was concluded, that the lack of heat treatment after each welding operation resulted in a low weld ductility and a material yield strength which was greater than the ultimate strength of the weld itself. Consequently, the new heat-treatment procedure was formulated.

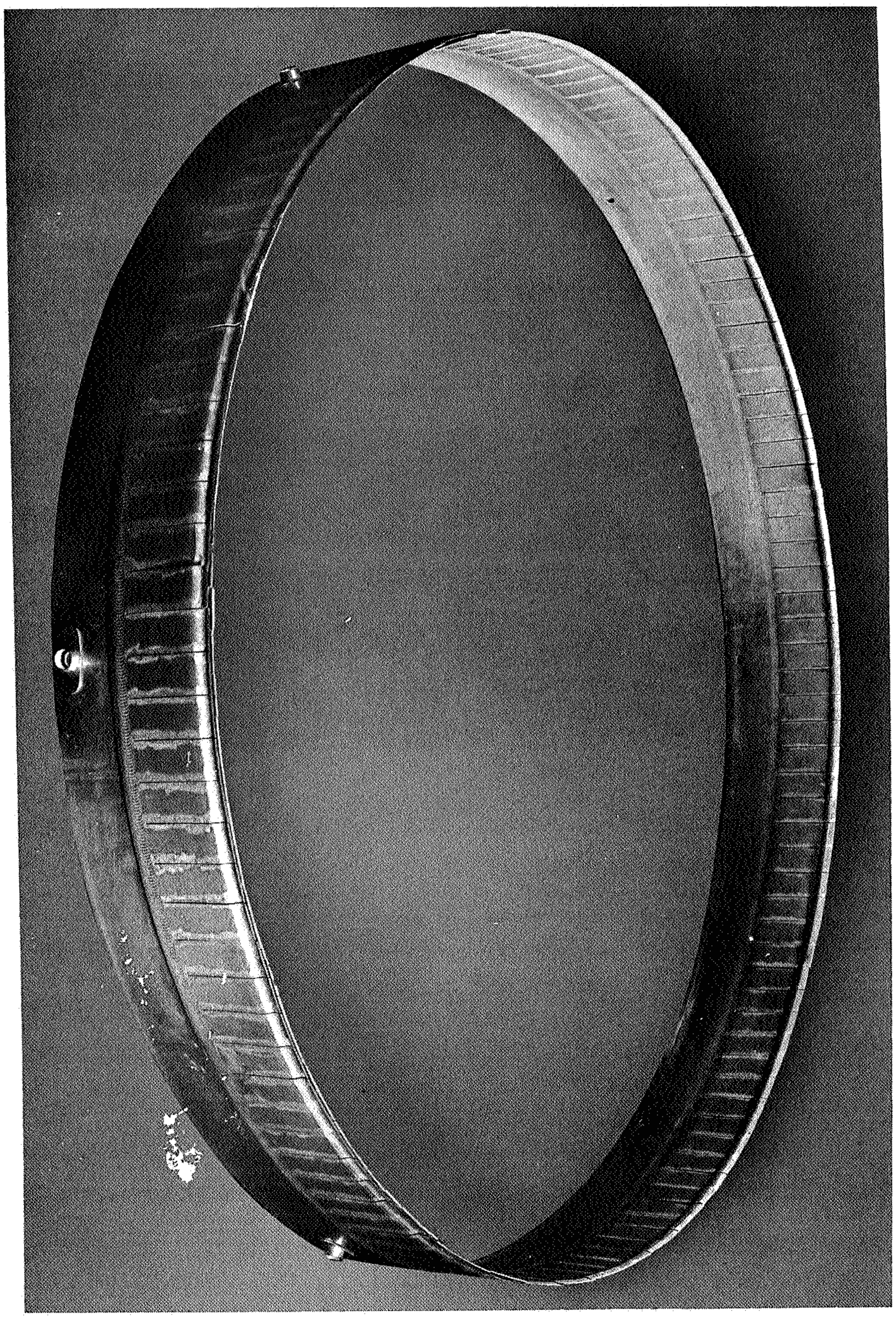

FIGURE 7-5. DOME END SEAL

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After the arc fusion welding step, the material is solution heat-treated at 995° F for 4 hours to the T42 condition; a cold water quench follows. The two rings are held under refrigeration until explosive forming and saw cutting are started. Next, the rings are offset to stagger the saw cuts and the pieces are resistance-welded together. The final step is an aging heat-treatment at 375° F for 36 hours to the T6 condition. This procedure has improved the weld strength and sheet metal ductility greatly, while introducing a very slight reduction in the mechanical properties of the material used.

Design Analysis. The peak bending stress, due to core lateral displacement at a 4-g acceleration, is 28,000 psi. The 21 psi pressure drop across the seal induces a bending stress of 25,000 psi, but this does not occur simultaneously with the 28,000 psi induced by acceleration. The stress due to the 0.030 inch radial shrink fit required to adopt the seal to the support plate is 8800 psi. These stresses are well within the yield strength which is over 50,000 psi.

Environment. The operating temperature of this seal is approximately 300° R and, as mentioned previously, the pressure differential between the reflector area and the core inlet plenum is approximately 21 psi.

C. FUEL CLUSTER ASSEMBLY

Design Philosophy. The fuel cluster is the typical fuel module which includes the necessary support hardware and coolant flow passages (Figure 7-6). The compressive loading of the fuel elements caused by the pressure drop is transmitted to the support block and then through the support washer, insulating cup, insulating washer, and support cone to the head of the tie rod. The tie rod carries the cluster load in tension, through the tie rod nut to the tie rod holder. The threaded area of the holder transfers the cluster load to the fuel cluster nut and the core support plate.

The coolant for the support plate attachment enters through slots in the holder, passes along the tie rod and exits between the preload spring coils. Tie rod coolant enters through four orifices in the cluster plate centering bushing, passes along the annulus between the tie rod and liner tube in the central element, and exits through the 12 holes in the support cone. From a plenum above the core elements, the fuel element coolant enters the 114 orifice jets in each cluster through holes in the cluster plate. The orifice jets are designed to provide a uniform exit temperature from each fuel element channel within the core by metering the flow in order to compensate for unequal flow channel resistance and heat generation. Each element channel mates with a hole in the support block through which the propellant exits from the core.

The tie rod extends approximately 5 inches above the element cluster, thereby providing flexibility for lateral movement of the cluster relative to its point of attachment to the support plate. The axial position of the elements relative to the tie rod holder, and therefore relative to the support plate in the core assembly, is adjusted by positioning the tie rod nut, which is then locked by a spring pin.

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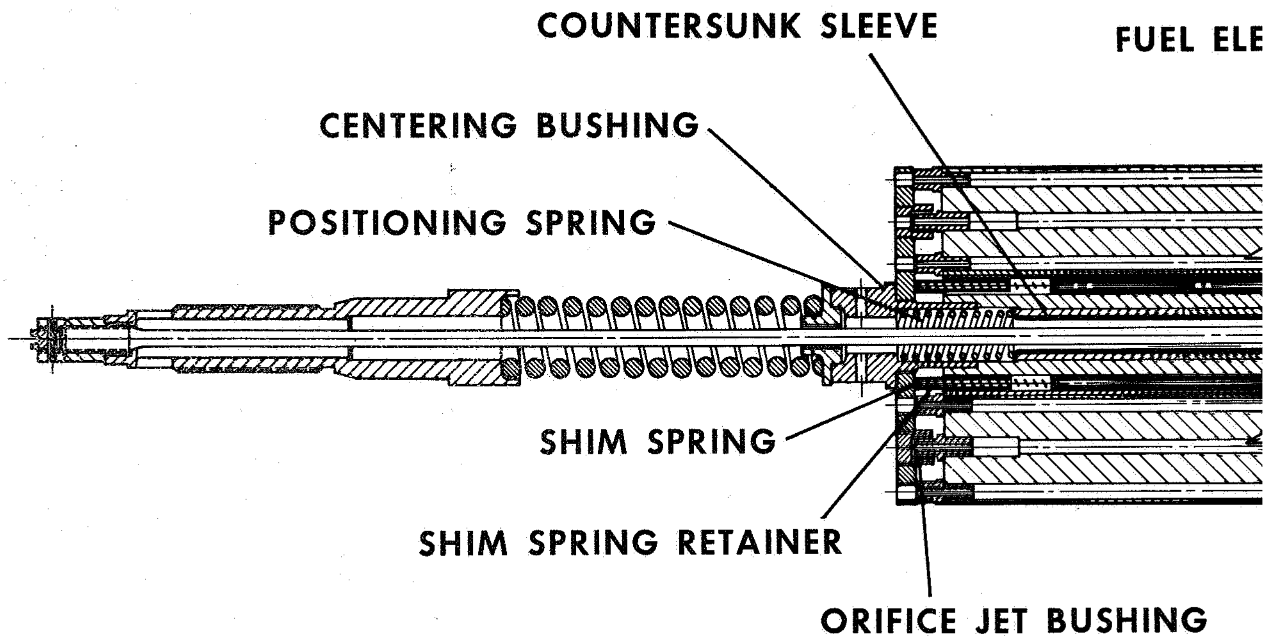

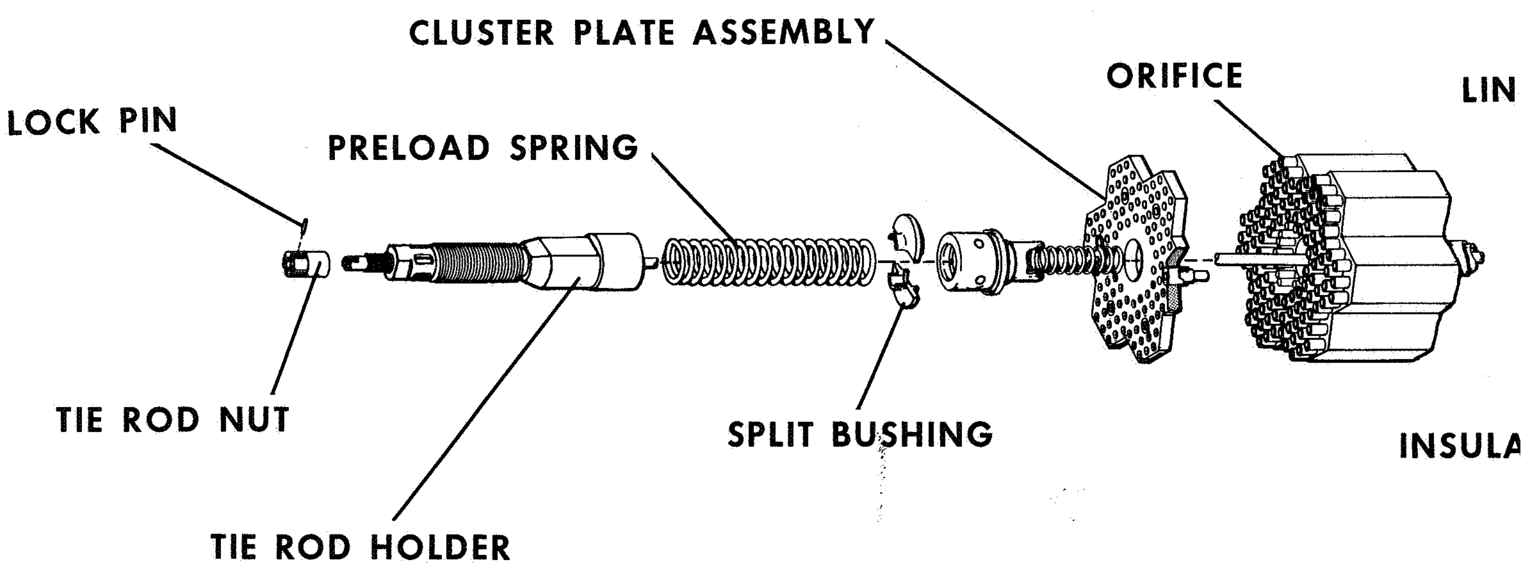

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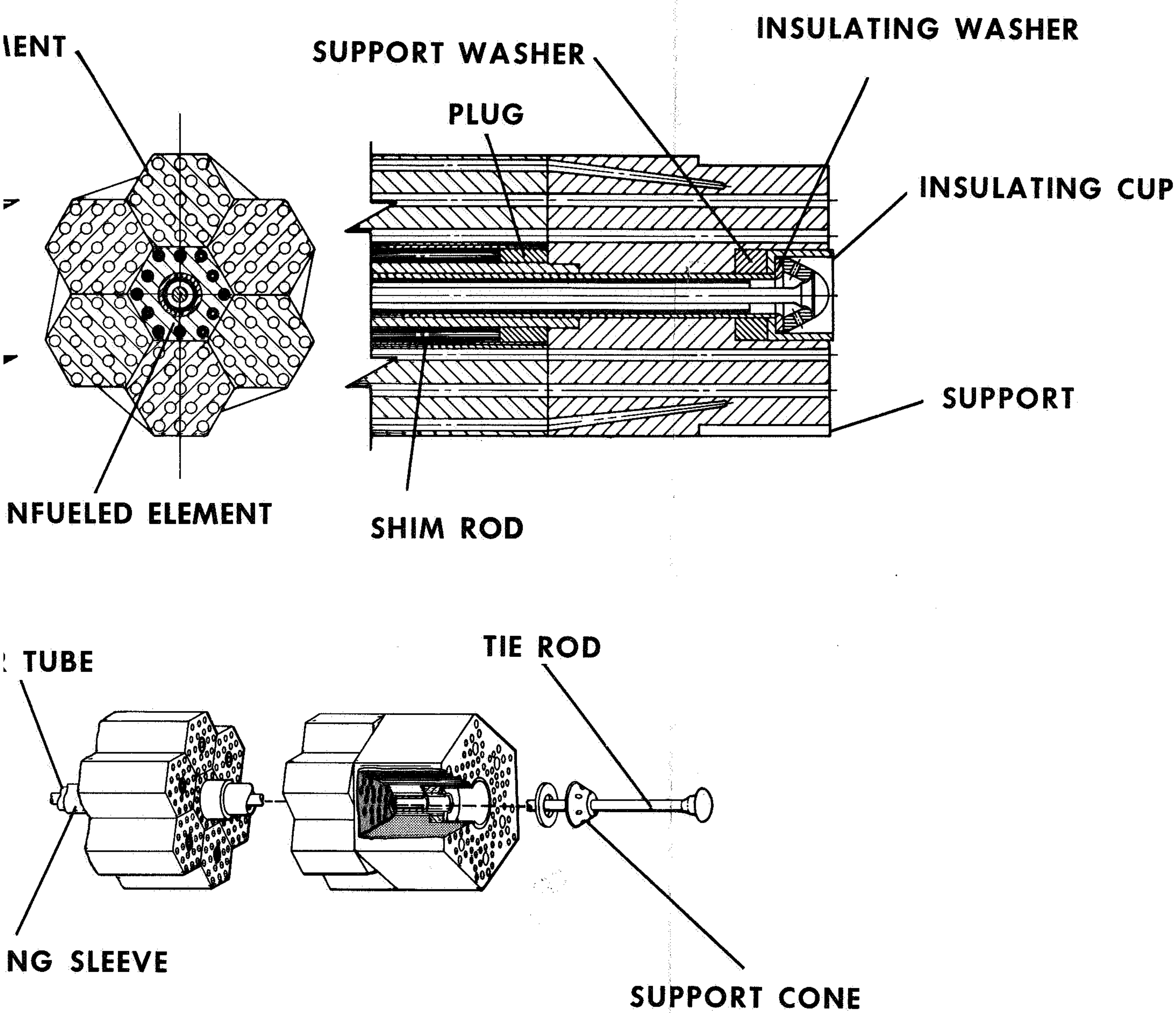

FIGURE 7-6. FUEL CLUSTER ASSEMBLY

Description. The fuel cluster assembly, which is 5 feet long, consists of a 52-inch long hexagonal central unfueled element, surrounded by six similarly-shaped fuel elements. This seven-element bundle is capped by a support block at one end and a cluster plate at the other; these units are loaded together by means of the preload spring and the tie rod. The regular fuel cluster assembly is shown in Figure 7-6, while Figure 7-7 shows the major hardware components which comprise the cluster. There are 199 clusters of this physical configuration, plus 42 irregular clusters in the peripheral area.

Seven different irregular configurations are used. Two configurations contain 3 central unfueled elements and tie rods which are distributed among the 15 fuel elements and the 1 partial element. Four other configurations have 2 central elements, 1 partial element, and either 7 or 8 fuel elements. The remaining cluster contains a single tie rod and central element, as in the regular cluster, but has only 5 fuel elements plus a partial element.

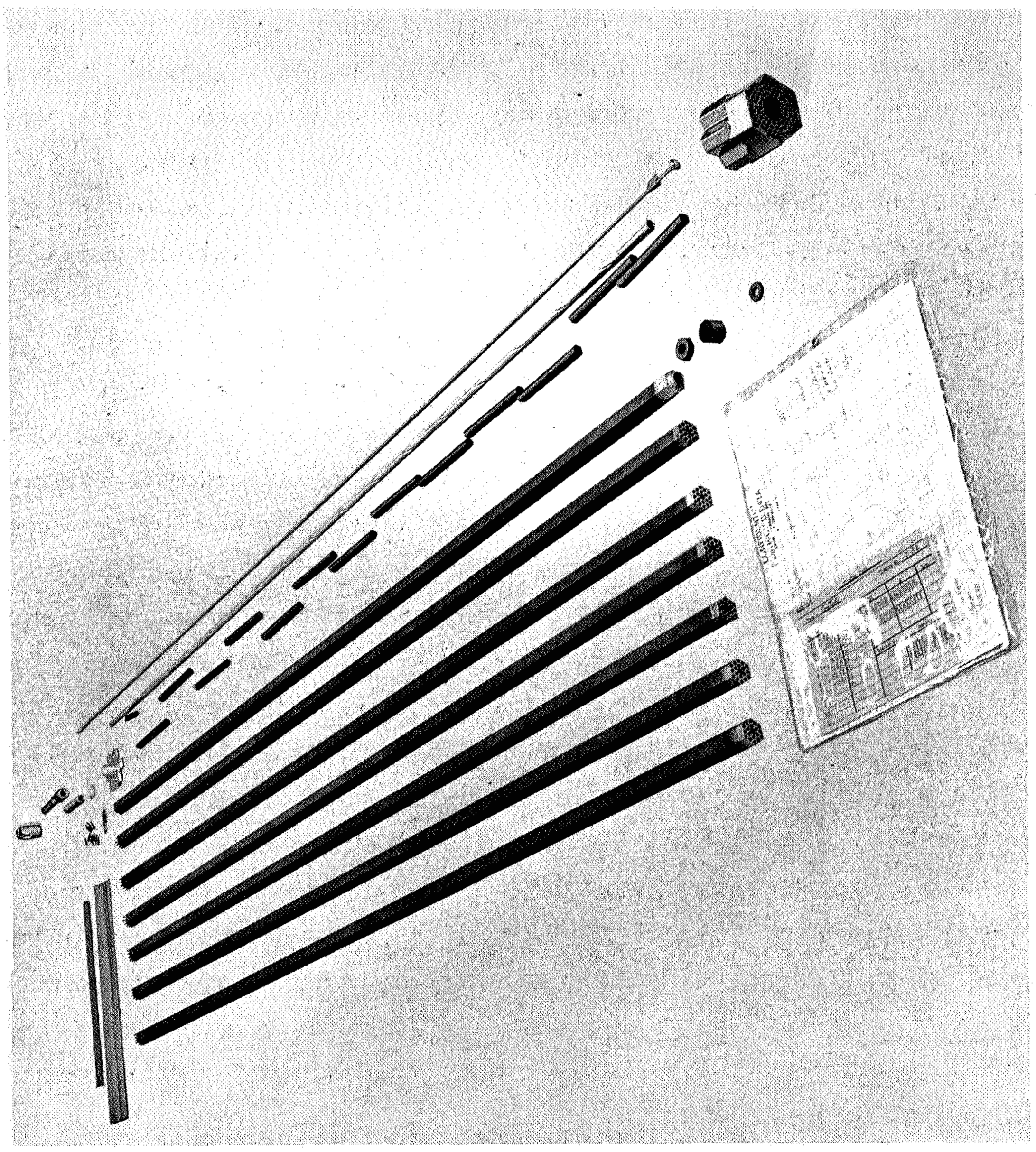

FIGURE 7-7. FUEL CLUSTER ASSEMBLY COMPONENTS

1. Liner Tube

Design Philosophy. The tube serves as a protective shield for the insulating sleeves to prevent flakes of pyrolytic-graphite from being carried by the hydrogen coolant to the support cone, where they could block the coolant flow passages. It also prevents the flow of hydrogen around the hot outside diameter of the insulating sleeve, thereby protecting the sleeves and the unfueled element from corosion. The helical groove in the liner tube serves as a centering feature to keep the tie rod from touching the wall of the liner tube where it could be in contact with the hot insulating sleeves. This provision eliminates the chance of a hot-spot developing on the tube wall and the tie rod. The groove also serves to damp out any flow-induced vibrations.

Description. The liner tube consists of a thin-walled tube (0.005 inch thick), 4-1/2 feet in length with one end flared out at 90 degrees to the centerline of the tube. The opposite or discharge end is tapered inwardly to eliminate the sharp circumferential lip which could damage the insulating sleeves during assembly and operation. The diameter at this point is reduced approximately 0.020 over the last 0.50 inch of length. The O.D. of the liner tube is approximately 0.25 inch, and a shallow helical groove is formed on a 3-inch pitch, which runs the length of the tube, beginning from a point 2 inches from each end. The tube has the flared lip at the inlet end. This lip rests on the chamfered upper end of the countersunk sleeve and is held against this sleeve by the positioning spring.

Material Selection. The liner tube is fabricated from Type 304 stainless steel tubing because of its good cryogenic ductility.

Design Analysis. The liner tube is subjected to a small and relatively constant tensile load. During assembly and before operation, it is in tension due to its own weight, and during operation it is in tension due to fluid friction. Both analytical and experimental results show that the flared lip will withstand the loading condition during operation.

Environment. The tube operates in a gaseous hydrogen environment at temperatures ranging from approximately -210° R at the inlet end to approximately 910° R at exhaust, with possible local hot spots to 1160° R.

2. Plate Assemblies

Design Philosophy. The compressive load of the preload spring (approximately 50 pounds) is transmitted to the elements and support blocks by means of the cluster plate assemblies. The orifice jets themselves, which protrude from the upper ends of the fuel elements, are loaded by the plate assemblies and, in turn, transmit this load to the elements and support blocks. The plate assembly bears on top of the orifice jets (19 per fuel element) which are cemented in the plate in each flow channel. The central orifice jet in each element is trapped by a bushing pressed into the cluster plate, and these bushings prevent the elements from falling out during handling, prior to core assembly. Flow holes located in the cluster plate direct coolant into each orifice jet but do not serve any flow-metering function. The centering bushing provides the orifices for metering tie rod coolant and, with the split bushing in place, centers the tie rod in the central element. The cluster plate assembly retains the elements against negative acceleration loading of the core during shipping, ground tests, or booster operation. The plates serve the added function of nuclear flux suppression for the core inlet area.

Description. Basically, each regular and irregular plate assembly consists of a cluster plate made of a neutron-absorbing material which has a number of centering bushings and orifice jet bushings pressed into it. The regular plate is the shape of the cross section of a seven-hex element bundle, with 114 flow holes and a large central hole for the centering bushing. The irregular plates take the shape of the element bundle in the irregular clusters. The bushings, which are made of aluminum, are pressed into the cluster plate at assembly. For every orifice jet beneath the plate assembly there is a flow hole approximately 0.1 inch in diameter in the plate. The shape of five of the peripheral cluster plates closely match five of the support block shapes. Two other plates are made identical to two of these plates since the corresponding element clusters are the same. This results in five different configurations for the 42 peripheral cluster plates.

To allow bundling of the fuel elements in an irregular cluster, only one of the centering bushings is pressed into the cluster plate. The one or two remaining centering bushings are free to shift in a clearance hole as the core elements are bundled.

Material Selection. The centering bushings and the orifice jet bushings are made of aluminum. Silver-indium-cadmium alloy (80, 15 and 5% weight, respectively) is currently specified for the cluster plate material. Stainless steel (300 series) containing 1% (by weight) gadolinium, is being investigated as a possible replacement because that material offers a higher temperature margin. A process for adding gadolinium-oxide powder to stainless steel powder and sintering the mixture, followed by hot rolling, is also evaluated in order to eliminate the problems involved with direct addition of gadolinium to the stainless steel.

Design Analysis. The following table lists important strength characteristics of the material used for this unit:

Characteristic	Specification Values (Minimum)	Test Sample 530° R	Test Sample 1060° R
Tensile yield strength (psi)	10,000	11,800	11,400
Ultimate yield strength (psi)	40,000	44,200	16,800
Elongation (%)	45	52	43

In addition, analyses have shown that in the improbable condition that the plate transmits the loads to the fuel element through the peripheral orifice jet without the benefit of any of the centrally-located orifices, the peak bending stress caused by the spring preload and the effect of the core lateral displacement resulting from a lateral force of 4g would be 7500 psi. A peak bending stress of 8400 psi has been calculated for the same assumption and for the loads resulting from the preload spring force, thermal expansion and pressure drops which are anticipated during operation. These stresses are within the yield strength of 10,000 psi and ultimate strength of 40,000 psi for the silver-indium-cadmium material used.

Environment. The cluster plate assemblies operate in a gaseous hydrogen atmosphere ranging from approximately 125° R during start-up to 300 to 350° R during operation. It is anticipated that the maximum temperature condition during transient operation and shut-down will not exceed 1000° R.

3. Preload Spring

Design Philosophy. The function of the preload spring is to provide an initial compressive load to the fuel cluster assembly components; it also serves to hold the cluster together but allow relative thermal expansion between the tie rod and fuel elements. The spring also serves to restrain the elements under negative accelerations due to handling, ground test, or booster operation. The spring force is transmitted to the cluster plate assemblies, to the orifice jets, through the fuel elements, to the support blocks, to the tie rod and finally to the tie rod holder.

Description. The preload spring is a simple helical compression spring which is made from 0.101 inch diameter wire wound to a 0.50 inch coil diameter with a free length of nearly 2.5 inches, both ends are closed and ground. The spring rate is approximately 136 lb./in. and, when deflected in the assembly position, provides a preload force of approximately 50 pounds to the fuel cluster assemblies. It is compressed between the tie rod holder and the split bushing.

Material Selection. Type 302 stainless steel spring wire was selected because it is relatively corrosion resistant and is satisfactory at cryogenic temperatures.

Design Analysis. Design calculations show the spring to be adequate for the intended application; the spring will prevent movement of the cluster up to approximately a 4-g negative steady acceleration loading.

Under normal preloading condition, a shear-stress of about 83,000 psi is anticipated for this spring. Furthermore, operating conditions could produce a stress of about 7800 psi. The extreme case of full compression of the spring produces a shear stress of 100,000 psi. However, all of these stresses are below the room temperature yield shear strength of 117,000 psi.

Environment. The preload spring operates in a gaseous hydrogen atmosphere at approximately 200 to 300° R.

4. Positioning Spring

Design Philosophy. The function of the positioning spring is to seat the liner tube against the chamfered end of the countersunk sleeve in the central element and to allow relative thermal expansion of the element to the insulating sleeves, while still providing a load to restrain the liner tube and sleeves during the negative acceleration which may occur during handling or booster operation.

Description. The positioning spring is a simple helical compression spring which is made from 0.025 inch diameter wire wound to a 0.35-inch diameter coil with a free length of nearly 1.5 inches; both ends are closed and ground. The spring rate is approximately 1.5 lb/in. and, when deflected in assembly, provides a spring force of approximately 0.5 pound to seat the liner tube against the chamfered end of the liner tube of the countersunk sleeve. The upper end is retained inside the centering bushing, while the lower end rests on the flared lip of the liner tube.

Material Selection. Type 302 stainless steel spring wire was selected because it is relatively corrosion resistant and is satisfactory at cryogenic temperatures.

Design Analysis. Calculations indicate this spring will retain the sleeves and liner tube with the cluster subjected to a 4 g loading.

Environment. During reactor operation, the spring is exposed to a temperature of approximately 300° R.

5. Orifice Jet

Design Philosophy. The orifice jet is a flow restriction device which meters the flow of hydrogen into each flow channel of the fuel elements. The orifice jet size is chosen to compensate for variations in effective element channel diameter and in predicted heat generation rate in order to achieve a nearly constant outlet temperature for all channels. The jets are cemented into the channel to help seal against bypass leakage and to ease handling of orificed elements. The protruding shoulder increases bearing area and helps seal against leakage.

Description. The orifice jet is approximately 0.375-inch long and 0.125-inch in diameter. The outside diameter is stepped at mid-length, where a protruding shoulder exists, while the bore diameter ranges from 0.0225 to 0.700 inch for the 12 different-sized orifice jets. The corner radius is tightly controlled at the inlet end of the central bore to provide a sharp edge orifice characteristic. At the inlet end, the O.D. is covered by a plastic cap which is color coded to identify the orifice jet size; this cap also protects the critical inlet corner during handling and installation. The orifice jet is cemented into the fuel element channel and, with the cap removed, is trapped between the cluster plate and the element.

Material Selection. A 300 series stainless steel was chosen for the orifice jets on the basis of corrosion resistance and high-temperature capabilities.

Design Analysis. Although analysis indicates that this part might experience local yielding because of thermal stresses, this condition will not prevent the part from functioning effectively.

Environment. Because heat generated in the fuel at the core inlet passes through the orifices to the coolant, temperatures as high as 500° F may occur during reactor operation.

6. Split Bushing

Design Philosophy. This bushing centers the tie rod in the centering bushing and thus in the central unfueled element. It also minimizes leakage between the centering bushing and the tie rod thereby channeling coolant into the metering holes in the centering bushing which would otherwise bypass the bushing. The split is required for a close fit to the tie rod because the head and threaded ends of the rod are significantly larger than the shank.

Description. The split bushing is a two-piece ring, split along a diameter, with a central hole and a reduced diameter shank which fits closely into a mating recess in the centering bushing. The cluster preload spring retains the split bushing in the centering bushing and around the tie rod. There is approximately 0.003 diametral clearance between the tie rod and the split bushing bore.

Material Selection. Aluminum was chosen for the split bushing because of its low nuclear heating and good thermal conductivity, and to be consistent with the mating centering bushing (in relation to the effect that thermal expansion exerts on the bushing).

Design Analysis. There is no significant loading on this part.

Environment. The split bushing is expected to range in temperature from 125° R at start-up to 300 to 350° R during operation. The maximum temperature expected during pulse cooling is approximately 1000° R.

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7. Tie Rod

Design Philosophy. The tie rods provide axial retention of the fuel cluster assemblies. Each regular fuel cluster is suspended from the core support plate by one tie rod. Either one, two, or three-tie rod clusters are provided in the core peripheral rows. The quantity of rods per cluster is based on the load-carrying capability of the support blocks and the rods. These tie rods must be capable of withstanding the core pressure drop, cluster preload spring force, and bending which is caused by lateral shock loading.

A single tie rod diameter is specified in the NRX-A regular and peripheral clusters, as opposed to tie rods of various diameters. In peripheral clusters, the specification of different tie rod diameters, related to their distance from the center of pressure loading, could result in an equal stress and elongation for all tie rods in a single cluster. Theoretically, a uniform loading would exist at the base of each fuel element, at its interface with the support block, but the load and stress at each cup counterbore of the support block would be different, some being higher than in the regular clusters.

The following statements explain why it was preferable to specify a single-diameter tie rod:

(1) A restoring moment at the base of the elements will counteract the unbalanced tie rod loading, thereby preventing the block from tilting.

(2) With no tilting, all tie rods will elongate equally and will carry the same stress and load. The support block loading and stress at the cup counterbores are, therefore, equal within a block and are less than those for the regular support blocks.

(3) Elimination of different-sized tie rods prevent errors in location of tie rods, simplifies identification, and thus improves reliability.

The 5-inch extension between the point at which the tie rods leave the core and the point of attachment in the support plate, in combination with the small diameter of the shank, provides the flexibility necessary to allow for the lateral shift which the core will experience as the result of lateral shock loads.

Description. The tie rod is a 0.110-diameter wire, approximately five feet long with a hot upset head at one end and a hot upset section at the other which is rolled to a No. 10-32 NF-3A thread.

Both of the formed sections blend into the 0.110-inch nominal shank diameter through generous radii. On the threaded end, there is an unthreaded extension with a transverse hole which is about 0.031-inch in diameter and a transverse external slot.

The tie rods pass through the support blocks, through the unfueled core elements, and extend approximately 5 inches beyond the core inlet, before attachment in the support plate. Of the 289 tie rods in the reactor, 90 are used in the peripheral clusters.

Material Selection. Inconel X-750 was chosen as the tie rod material because it possesses the necessary strength characteristics, because it has no ductile-brittle transition in the operating temperature ranges which will be encountered, and because there is more knowledge about its characteristics in a radiation environment than there is about other materials which could have been tried.

Inconel 718 is being investigated as a back-up material because it has greater strength, ductility, and temperature capability. Consequently, it allows a greater margin for increased design requirements. However, little is known about the behavior of this material in a radiation field; therefore, tests must be completed before a material change can be made.

Design Analysis. Considering the nominal axial clearance of 0.095 inch between supports the exit end temperature of a tie rod may be 530° R hotter or colder than the tie rods in adjacent clusters before the interlocks on the support blocks contact. This value is based on a linearly increasing tie rod temperature difference which exists between core inlet and core exit. The difference is reduced to 280° R when the extreme tolerance on all parts is considered and the axial clearance is reduced to 0.050 inch. The maximum nominal exit temperature variation of the tie rods in any two adjacent clusters is expected to be about 65° R. This temperature difference is caused by power variation. This maximum permissible tie rod operating temperature is 1660° R for Inconel X750; this limit is based on ductility reduction at temperatures above this value. The maximum nominal operating temperature expected in the tie rod is 900° R, which increases to 1200° R under the anticipated decay heat condition.

When the tie rods are subjected to an axial load that consists of a core pressure drop of 129 psi (plus 6% margin) and pre-load spring forces at operating temperature, the stress in the tie rods in the peripheral clusters varies from 42,000 psi to 55,000 psi, compared to 58,000 psi for those in the regular clusters. The maximum bending stresses due to the core eccentricity are expected to occur when the core is not operating; these stresses have been calculated as 65,000 psi. While the minimum specified yield strength is 80,000 psi, sample testing of tie rods indicated that actual values are between 98,000 and 104,000 psi.

Environment. The tie rods must operate in a radiation field at temperatures ranging from approximately 200° R to 900° R. The spiral ridge formed on the inside of the liner tube (through which the tie rod passes) keeps the rod centered and prevents hot spot conditions due to thermal bowing of the rod.

8. Tie Rod Holder

Design Philosophy. The holder transmits the force of the preload spring to the tie rod nut, but allows coolant flow to pass through elongated holes between the holder I. D. and the tie rod, and through the preload spring coils. The holder also transmits the full cluster pressure load through the threaded area into the fuel cluster nut and then to the support plate. During installation of the cluster nut, it is prevented from rotating by the flats which fit into a broached slot in the support plate.

Description. The tie rod holder is a threaded tubular component about 0.5 inch in diameter and 2.75 inches in length. It has a shoulder at one end and six elongated holes at the other end. Two flats are milled into the cylindrical body of the holder which is positioned against the tie rod nut by the cluster preload spring. In the core assembly, the holder is pulled into a bore in the support plate by the fuel cluster nut.

Material Selection. Type 303 SE stainless steel was chosen for the holder because it has good machinability (relative to the other class 300 steels) and high ductility at low temperatures. The threaded area is hardened to alleviate possible galling.

Design Analysis. The stresses due to fuel cluster loading are basically compressive and in the order of 23,000 psi maximum at the flow hole area. The yield strength for the material is 65,000 to 100,000 psi, depending on the diameter of the raw material.

Environment. The operating temperature of the tie rod holder is about 650° R; it may rise as high as 1050° R during decay heating.

9. Fuel Element

Design Philosophy. The fuel elements, which are non-structural components, act as the heat generating energy medium of the reactor. While these elements are maintained chiefly in compression due to the axial pressure and the lateral support loading, they do have tensile stresses induced by thermal gradients and longitudinal bending (maximum at the core periphery) to conform to the thermally induced diameter variation of the core.

In addition to the full fuel elements, partial elements are used in the core periphery to provide the transition from the irregular periphery to a more circular shape.

Although the fuel elements are assembled in clusters of six (plus one unfueled central element), once they are in the core assembly, they act as a tight bundle of elements. The support blocks and cluster plates provide only axial restraint, they do not position the elements laterally.

The elements of most regular clusters have a single value of nominal fuel loading. However, at the periphery, there are six nominal loading values that are used to level off the nuclear flux and the heat generation peaking which would otherwise occur in the area near the reflector. In an attempt to provide the most uniform heat generation profile, elements within a given nominal loading are selected for exact position in the core on the basis of actual measured fuel loading.

Description. The fuel element is 52 inches long with a hexagonal cross section which is 0.753 inches across flats. It contains 19 full-length axial flow holes, which are nominally 0.0968 inch in diameter at inlet, with a 0.001 inch diametral decrease from flow inlet to exit due to an increase in coating thickness. For a 0.75 inch length at the flow exit end, the hexagonal external surfaces are stepped down by approximately 0.004 inch on all six faces. The coolant flow holes, the flow exit face, and the stepped-down hexagon length of the fuel elements are coated with niobium carbide for corrosion protection. At the central flow hole inlet end there is a 0.5 inch deep by 0.1 inch diameter counter bore which is accurately centered in the hex cross section.

The partial fuel element is identical to the full element except that two rows of flow holes (7 holes) have been removed by a cut parallel to one of the hex faces.

Material Selection. The fuel material consists of beads of enriched uranium dicarbide particles, approximately 100 microns in diameter, including a 25-micron pyrolytic carbon coating. These micro spheres are uniformly dispersed in a graphite matrix. The element configuration is extruded to shape with some excess material in the 19 flow holes and on the hexagon surfaces. The pyrolytic coating reduces hydrolysis and dimensional instability of the fuel while the graphite matrix is a nuclear moderator which is capable of withstanding the extreme temperatures involved.

Design Analysis. Stresses in the fuel elements arise from several different conditions:

(1) The compressive stress in the element due to the flow pressure drop varies from 130 psi at the inlet end to 190 psi at the exit end.

(2) The compressive stress due to the preload spring is about 30 psi.

(3) The maximum tensile stress due to the longitudinal bending of the outer fuel elements is less than 300 psi; this maximum stress occurs at a point 42 inches from the inlet end of the core.

(4) Because the fuel element is bundled against the surrounding elements, differential expansion due to variation in the operating temperature of the fuel elements will cause stresses in the element. The magnitude of these stresses depends both on the temperature difference between the elements and the shear force caused by friction between the elements. This factor produces a maximum tensile stress of 800 psi, which occurs in the axial center of the core.

(5) Thermal stresses result from three different temperature gradients. A thermal gradient of 160°R is required to conduct the heat out of the fuel element at the power peak in the center of the core. The gradient produces a local tensile stress of 500 psi at the coolant hole wall. A second thermal gradient will occur between fuel elements operating at different temperatures. This gradient leads to a maximum tensile stress of 150 psi. Lastly, if there is coolant leakage through the filler strips at the core periphery, there can be a thermal gradient in the outer fuel elements of as much as 500°R. This gradient could cause a localized tensile stress of about 1500 psi. Experiments have shown that this leakage does not cause surface cracking of the elements.

Because these stresses occur at different points of the element and at varying volumetric points, a Weibull analysis was conducted. The analysis shows that the probability of survival for the most highly stressed interior element is higher than 0.998. On the basis of this stress distribution, a probability of survival for the whole core can be estimated. This probability of

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survival (probability that no elements in the core will break because of this stress distribution) is about 0.98. This indicates one fuel element failure in approximately 50 cores. It should be noted that probability of survival does not consider thermal gradients in the peripheral fuel elements.

Environment. The elements operate at temperatures between 1000 and 4450° R, with maximum local gradients limited to approximately 500° R. The axial loading due to pressure and spring preload is about 80 pounds for a 130 psi pressure drop across the core.

10. Unfueled Element

Design Philosophy. The central hole of the unfueled elements is used to provide a cooled passage for the tie rods and to hold the tubular insulating sleeves. By means of a counterbore at the flow inlet end of the element, the cluster plates are positioned relative to the core element bundle. In a similar manner at the opposite end, a protrusion mates with the support block counterbore to locate the support blocks relative to the element bundle.

The multi-hole elements also carry the moderator or the poison nuclear shims which are used locally to adjust the reactivity of the core in order to avoid hot spots, and overall to adjust total reactivity.

Description. The unfueled elements are 52 inches long with a hexagonal cross section which is 0.753 inch across flats. At the flow exit end of the core, these elements have a 0.5-inch diameter protrusion which is just under 0.25 inch long. For a 0.75-inch length at the same end, the hexagonal surfaces are stepped down by approximately 0.004 inch on all six faces. This is done to provide for dimensional increases due to coating. There are two types of unfueled central elements: one contains only a 0.375 inch diameter central bore and is used in the irregular clusters; the other, which is used in all regular clusters, has in addition to the central bore, twelve 0.1-inch diameter, full-length holes which accept the nuclear shims. These holes are blocked at the flow exit end with plugs which are 0.375 inch in length. This end of the element includes a niobium carbide case which locks the plugs in place and provides corrosion protection for both the stepped-down hex and a 3.25-inch length of the central bore.

Material Selection. The unfueled element material is extruded, high-purity graphite, which is the same as that used for the matrix in the fueled elements. The graphite is a nuclear moderator which is capable of operation at the extreme temperatures which will be encountered.

Design Analysis. The stresses in the unfueled element are similar to those in the fueled element, with the exception that there can be no temperature gradients due to the core peripheral leakage and the heat generation is much lower.

Additional stresses exist in the bottom of the 12-hole unfueled element; these stresses are caused by the 130 psi pressure differential between the plugged holes and the core exit pressure. This additional stress is about 500 psi at the exit end of the element, decreasing to 0 at the inlet end. As a result of these conditions, the probability of survival for unfueled elements is higher than for fueled elements.

Environment. These elements are subjected to temperatures ranging from 1000° to 4450° R.

11. Keyed Element

Design Philosophy. The cemented keys retain the filler blocks and the core band in the proper axial position relative to the elements. With the core band in place, an added safety margin is achieved because the band will hold an individual filler block in place if a key should fail. If the key became unbonded, it would be trapped in the recess of the filler block.

Description. The keyed element assembly consists of a fuel element or a partial element; both have a key which is cemented to a longitudinal face of the element. The key is centered on the element face width, with the end located just over 1 inch from the flow inlet end of the element.

The key is 1/4 inch wide, projects about 0.125 inch from the element surface, and is 2.2 inches long. When installed in the core assembly, the key portion of the assembly fits into a recess in the filler block.

All of the 42 partial elements have a key. In addition, 108 of the full elements have one key while 6 elements have 2 keys.

Material Selection. The key is made of the same extruded graphite as that used for the extruded fuel element matrix in order to minimize the different expansion between bonded parts. Great Lakes Carbon P-514 graphite cement has been chosen as the bonding agent.

Design Analysis. The loading of the key bond area averages about 10 pounds shear force, with a maximum of 14 pounds (26 psi) at the largest filler blocks. Laboratory tests have demonstrated that the keys have a minimum bond shear strength of 210 psi (113 pounds).

<u>Environment.</u> The key portion of the assembly operates at about 1000° R. If the friction force between the filler block and element is neglected, then the keys will transfer the filler block loading (20 psi pressure drop across the block) through shear loading of the cement bond. The key shear load is in the direction opposite to the pressure load on the element.

12. Nuclear Shims

Design Philosophy. The nuclear shim rods adjust the reactivity of the core so that the control drums can effectively start or stop the power generation. The shim rods permit local and general adjustment of the nuclear power generation by insertion of plain graphite or nuclear poison bearing shims. The shims are placed in the 12 holes in the central elements of the 199 regular core clusters. All multi-hole element shim holes are loaded with either four graphite or five poison shims.

Description. The nuclear shims are long, slender cylinders, 0.088 inch in diameter, with a 0.030-inch diameter central hole. The graphite moderator shims are about 12.75 inches long, while the poison shims are just over 10 inches long. These shims are located in the twelve 0.100-inch diameter holes in the multihole unfueled element; they are compressed between the element plug at the core exit end and the shim spring at the inlet end.

Material Selection. The moderator shims are extruded, high-purity graphite, while the poison shims are extruded with 66% (by weight) of tantalum dispersed in the graphite matrix in the form of tantalum carbide.

Design Analysis. There is no significant loading on this component.

Environment. The shims are subjected to temperatures ranging from 1050° to 4450° R, and are held in position by the force of the shim spring, which is just over 0.1 pound.

13. Support Blocks

Design Philosophy. The core support blocks retain the core elements axially in reaction to the preload spring in the cluster assembly and the pressure loading during reactor operation. Each block is slightly smaller than the cluster of elements it retains, so that the elements may be tightly bundled. The smaller blocks also allow limited axial motion between adjacent clusters without interference (Figure 7-8).

The poison wires, which pass through the coolant flow holes of the block and elements, provide alignment while the cluster is being assembled into the core. These wires are removed prior to operating the reactor. Then, misalignment of flow holes in the core assembly due to rotation of the block relative to the fuel elements is limited by the interaction among adjacent support blocks.

The interlocking feature gives the block the ability to retain the cluster in place. Should a single tie rod fail, the interlock transfers the cluster load through those of the six adjacent blocks to their tie rods after a limited axial travel of the clusters.

Description. The support blocks,which are located at the nozzle end of the core, are 2.25 inches deep and have a dual-shaped peripheral contour, with an abrupt transition between the two near the mid-depth point. The inlet end takes the shape of the adjacent seven hex bundle of elements (for the regular blocks), while the exhaust end is hexagonal in shape and rotated relative to the 7-hex shape. This configuration provides interlocking ledges, with 0.093 inch nominal axial clearance between adjacent blocks. The regular blocks contain 114 flow holes which are approximately 0.1 inch in diameter. These holes are fed from the fuel elements. A central hole just under 0.375 inch in diameter is counterbored at both faces. The inlet face counterbore receives and locates a projection of the central element, while the other counterbore on the hexagonal end receives the support washer and insulating cup. Because of the transition in periphery shape, it is necessary to provide a one-degree angle on 12 of the flow holes in order to keep the holes from breaking out through the edge of the block. In addition, 12 other holes must be drilled at an 8-degree angle, each joining one of the straight-through holes in an enlarged hole which is just over 0.125 inch in diameter.

The core periphery support blocks are similar to the regular blocks, except that they are designed to fit the shape of the irregular fuel clusters. There are seven different configurations used for the 42 peripheral support blocks. Two of these blocks contain provisions for three central element (and tie rod) locations, four accept two central elements, and one block accepts only a single unfueled element.

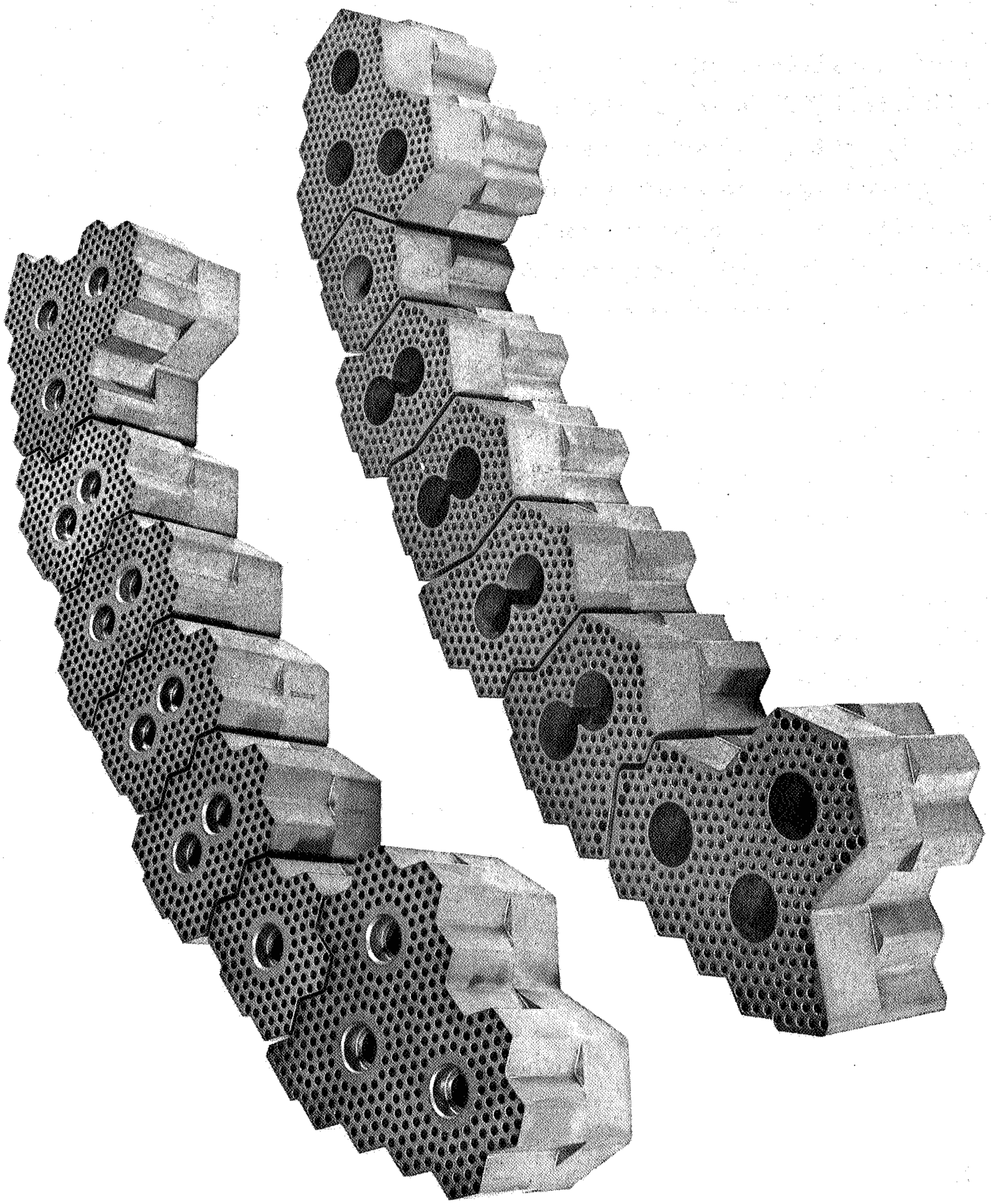

FIGURE 7-8. PERIPHERAL SUPPORT BLOCKS

Material Selection. Grade ATJ graphite was selected for the block material on the basis of good strength, a statistically meaningful quantity of available data on properties, and a match of thermal expansion coefficient to that of the fuel elements. All surfaces of the block which may be exposed to high temperature hydrogen flow are protected by a 0.002 - to 0.005-inch thick case of niobium carbide which was selected to prevent corrosion.

Design Analysis. Experimental testing has shown an average strength in the order of 7 or 8 times the design load of about 500 pounds. This safety margin is necessary for higher reliability because the graphite parts have a wide variation in strength and because thermal stresses which are superimposed during operation create a stress of 640 psi.

Environment. During reactor operation, regular support blocks carry the axial fuel cluster load of approximately 500 pounds at a temperature of 4150° R.

14. Support Cone

Design Philosophy. The cone transfers the pressure loading of the elements plus the load of the preload spring from the support block to the tie rod head. The 12 holes allow passage of tie rod coolant flow which also cools the cone itself. The conical shape allows a transition from a large bearing area, to the small diameter shank of the tie rod. The loading of the support block is improved by this increase in bearing area.

Description. The support cone is approximately 0.5 inch in diameter by 0.25 inch long. It is perforated by 12 holes which are just under 0.068 inch in diameter. The cone is located inside of the insulating cup, between the tie rod head (hot end) and the insulating washer.

Material Selection. A molybdenum alloy was chosen for the cone material because of the high temperature margin and the excellent thermal conductivity which reduces stress and temperature.

Design Analysis. The tie rod load coupled with the reaction at the base of the cone tends to turn the cone inside out. This condition creates a maximum stress of less than 45,000 psi in the cone, which compares favorably against the yield strength which is above 75,000 at operating condition. Also, there is an axial compressive stress of 12.000 psi due to the maximum tie rod load.

Test results show that a typical support cone can withstand over one million cycles at a vibratory load level of 700 pounds, ±250 pounds, and a temperature of -235° F. These cones have also withstood loads in excess of 4500 pounds during the ambient static suppor block tests.

Environment. The cone carries the full loading of the cluster (500 or 600 pounds) at temperatures ranging from room temperature to 800° R, with local values to 950° R in contact with the insulating washer.

15. Insulating Cup

Design Philosophy. The insulating cup separates the relatively cold tie rod coolant, exiting from the cone, from the hot support block. It prevents excessive thermal stresses from occurring in the critical support blocks and reduces radiant heating of the tie rod head.

Description. The cup is approximately 0.75 inch in diameter, 0.5 inch long, and 0.060 inch thick. It is coated on both the I.D. and O.D. along the 0.25 inch from the open end with 0.001- to 0.002-inch niobium carbide. The cup, which is recessed into a counter-bore in the downstream end of the support block, surrounds the support cone and tie rod head.

Material Selection. The cup is made of pyrolytic graphite which is deposited in a cup shape and machined to final size. This material provides the high thermal insulation required, while remaining structurally sound at extreme temperature conditions.

Design Analysis. Analysis shows that the maximum stresses occur during start-up. Since it is assumed that the base of the cup will be kept from expanding by the support core load, while the rest of the cup will expand freely, the maximum bending stress anticipated is 6400 psi. This stress will occur in the outer layer of the insulating cup at the point where the base joins the cylinder. The tensile strength of the pyrolytic graphite is above 10,000 psi at this temperature.

Environment. The outer portion of the cup, which is in contact with the support block, is expected to reach about 4150°R, with an operating gradient of about 3500°R through the cup wall.

16. Support Washer

Design Philosophy. The support washer transfers the fuel cluster loading from the support block to the insulating cup and then through the insulating washer to the support cone. The support washer distributes the reaction load of the cone and cup to improve the loading of the support block and increase the load carrying capacity of the support system.

Description. The washer is 0.25 inch thick, has an O. D. just under 0.75 inch, and an I. D. of 0.375 inch. It is placed into the support block counterbore before the insulating cup, and is sandwiched between the cup and the block.

Material Selection. This washer is machined from pyrolytic graphite plate so that the plane of deposition is parallel to the flat faces of the washer. Pyrolytic graphite provides the strength required to meet any non-uniform loading conditions which might be encountered at the extremely high operating temperature.

Design Analysis. The gross loading of the support washer results in an average compressive stress of 1600 psi, with a local bearing stress ranging as high as 2100 psi. The design strength for this item is above 8000 psi.

Environment. The support washer is subjected to a fairly uniform temperature of approximately 3950°R during reactor operation.

17. Shim Spring

Design Philosophy. This spring positions the nuclear shims in the multihole elements while still allowing for relative thermal expansion between the elements and the shims.

Description. The shim spring is a helical compression spring composed of an 0.009-inch diameter wire with a coil diameter just under 0.062 inch and a free length of 1.047 inches. The spring constant is approximately 2.4 lb./in. The unit is compressed between the nuclear shims and the cluster plate, through the shim spring cap.

Material Selection. Inconel X has been selected as the material for this spring because it supplies a high temperature resistance.

Design Analysis. The maximum calculated stress, which is caused by extreme tolerance build-up plus a superimposed 4-g inertia loading, is approximately 30,000 psi, while the room temperature yield strength of the spring material is 40,000 psi.

Environment. Operating temperature of the spring is about $1000^{\circ}R$. The compressed load at assembly is slightly over 0.1 of a pound.

18. Shim Spring Retainer

Design Philosophy. The shim spring retainer centers the spring in the shim hole, prevents binding on the edge of the hole, and traps any graphite particles that may be generated within the shim hole, preventing them from reaching critical areas of the core.

Description. The shim spring cap is a small 0.010 inch wall tubular part, 0.090 inch in diameter, and 0.630 inch long. Except for a 0.040 inch diameter hole in the center, one end is closed. The cap surrounds the shim spring and is compressed against the cluster plate by the spring.

Material Selection. The cap is machined from type 304 stainless steel.

Design Analysis. There is no significant loading on this component.

Environment. The shim spring retainer operates at about 800°R, with the temperature rising to about 1000°R during pulse cooling.

19. Insulating and Countersunk Sleeves

Design Philosophy. These sleeves act as thermal insulation to prevent excessive thermal gradients in the unfueled elements; they also limit heat flow into the tie rod coolant channel.

Description. The insulating sleeves are 4 inch long tubes with a 0.25 inch I.D., and a wall thickness just under 0.060 inch. They are sandwiched between the unfueled central element and the liner tube along the full length of the element. The countersunk sleeve which is only 1.350 inches long is used at the inlet end to accommodate the flared end of the liner tube.

Material Selection. The sleeves are machined from pyrolytic graphite which is deposited on a cylindrical mandrel. Pyrolytic graphite is required because of the extreme temperatures and the low thermal conductivity requirement.

Design Analysis. The temperature gradient (which is in excess of 3200° R) across the insulating sleeves causes a hoop stress of 6400 psi at the inner surface of the tube. The strength of the pyrolytic graphite is about 7000 psi at the inside surface temperature.

Environment. The O.D. of the sleeves is subjected to the unfueled element temperature which can be as high as 4450° R, while the I.D. (at other locations) can get as cold as 200° R.

20. Insulating Washer

Design Philosophy. This washer supports the column of insulating sleeves against axial movement, and the I.D. blend radius provides a fairing for flow exiting from the tie rod channel. The insulating washer also provides additional insulation at the contact surface of the support cone.

Description. The insulating washer is 0.071 inch thick by 0.609 inch in diameter, with a full blend radius at one face of the 0.24 inch diameter central hole. The washer is inserted into the insulating cup and is compressed between the support cone and the cup, with the I.D. blend radius opening toward the cone.

Material Selection. This component is machined from pyrolytic graphite which is deposited in plate form, so that the flat faces of the washer are parallel to the plane of deposition. Therefore, the material provides excellent thermal insulation in the direction normal to these faces.

Design Analysis. The thermal stress induced in the washer, if restrained from bowing, is approximately 1600 psi, compared to a 7000 psi estimated mean strength.

Environment. The temperature of the washer ranges from about 1000° R, in contact with the support cone, to 1950° R in contact with the insulating cup.

D. FILLER STRIP ASSEMBLIES

Design Philosophy. The filler strips (Figure 7-9) provide the transition from the irregular shape formed by the hexagonal core elements to the cylindrical section required to prevent binding during a lateral shift of the core. The strips simplify sealing between the core periphery and adjacent components, retain the insulating tiles by sandwiching them against the core, and provide a ledge for axial retention. The strips are loaded against the insulating tile and core elements by the seal segments of the lateral support system. The regular geometric shape of the core periphery leakage path allows better definition of the core periphery cooling and pressure distribution.

The local projections at the nozzle end of the filler strips provide axial retention of the strips within the lateral support system graphite barrel.

Description. The filler strip assemblies have an irregular cross section which is approximately 3/4 inch in width. These assemblies form the transition from the hexagonal element bundle to the circular core periphery. Each assembly is about 49 inches long, with a 1-5/16 inch length ledge located about 4 inches from the nozzle end of the strip. The ledge projects 0.150 inch from the radius contoured periphery of the filler strip. On the opposite side of the strip, there are one or two columns of insulating tiles, each column containing 15 tiles. There are 27 different shaped filler strips, and since there are 6 strips of each shape, the core assembly has a total of 162 filler strips. Of the 27 shapes, 15 have a single column of tiles, while 12 have a double column. The nozzle end tiles are recessed into the strip by about 1/16 inch. Each insulating tile is cemented to the filler strip between two protruding ledges which are on the longitudinal ends of the tile.

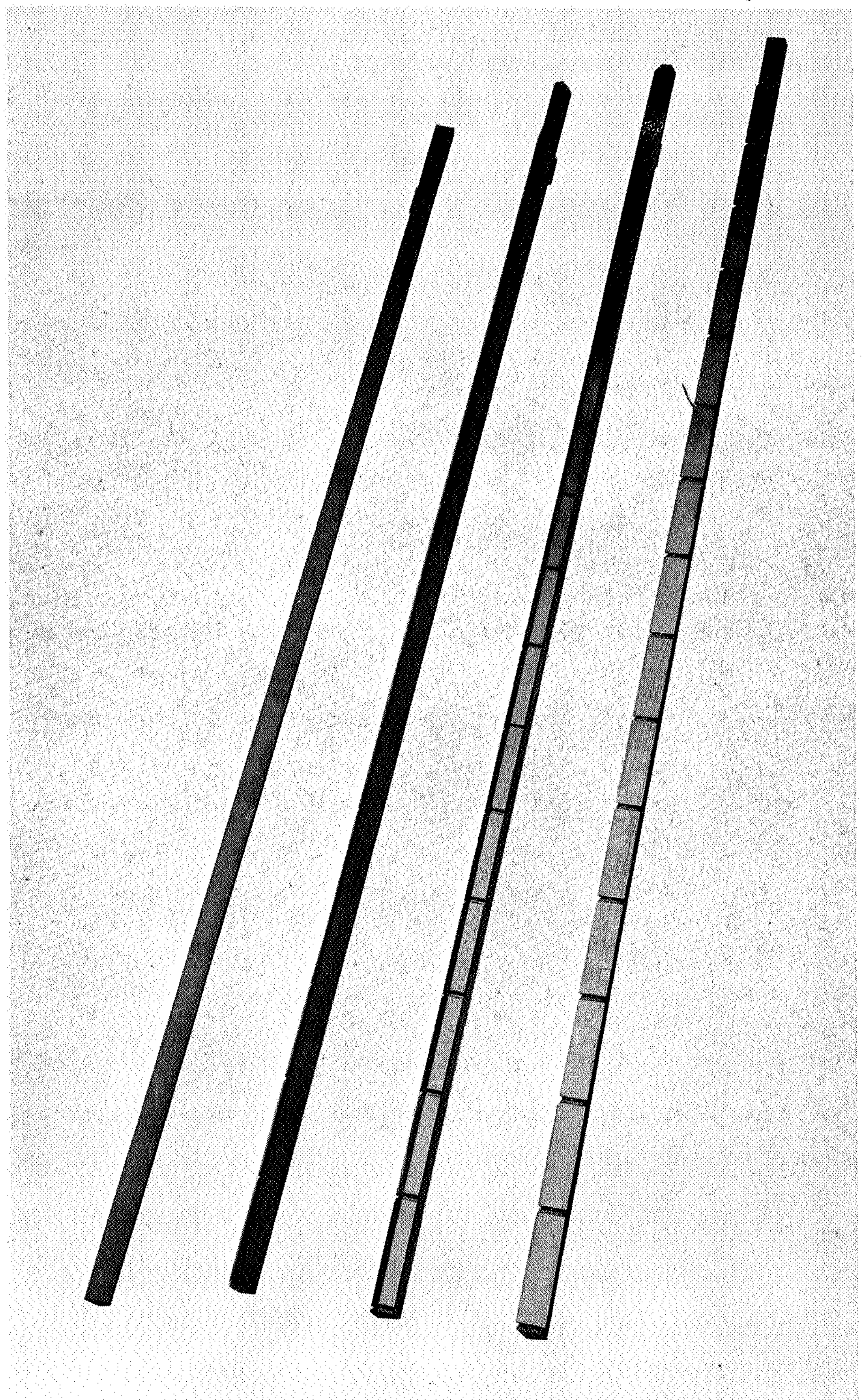

FIGURE 7-9. FILLER STRIP ASSEMBLY

Material Selection. Extruded Graphitite G was chosen for the filler strip material on the basis of strength density (for nuclear consideration), and thermal shock resistance.

Design Analysis. Six distinct factors operate to produce stresses in the filler strips:

1. Bending of the filler strips to follow the thermal curvature of the core. The bending produces a maximum tensile stress of 320 psi in the largest filler strip and 100 psi in the smallest.
2. Stresses caused by radial temperature gradients across the filler strips. The maximum radial temperature gradient is 140° R, yielding a maximum tensile stress of 210 psi in the largest filler strip and 175 psi in the smallest.
3. Friction shear forces which produce an axial stress in the filler strip if there is motion of the core relative to the filler strip. The maximum axial tensile stress caused by this friction force is 250 psi in the largest filler strip and 550 psi in the smallest.
4. The axial pressure drop produces a compressive stress of 129 psi.
5. Local bending stresses of less than 650 psi are produced at the base of the filler ledges caused by the expansion of the insulating tile relative to the filler strips.
6. Because the filler ledge is nearer the core than the rest of the filler strip, the ledge will act as a local hot spot on the filler strip. A maximum tensile stress of 840 psi may result locally at the base of the filler ledge due to the thermal gradient.

Taking all of these factors into consideration, the probability of survival for the largest filler strips has been computed as 0.9998, while the smallest strips has been computed as 0.99994.

Environment. During operation, the strips run between 1000°R and 4450°R.

1. Insulating Tile

Design Philosophy. Because of its high thermal resistance, the insulating tile helps to prevent excessive thermal gradients and resulting thermal stresses from occurring in the fuel elements and filler strips, particularly during start-up and shut-down transients. The decision as to where the insulating tile was to be placed was based on the fact that differences in thermal and nuclear characteristics of the fuel and the filler strips would result in different heat generation and conduction rates and, therefore, significant temperature differences in the two areas during transients.

The tile edges are beveled to form mitred joints for most locations. At points where mismatch occurs between tiles on adjacent filler strips, the bevel angle is changed to prevent more than a line contact between the two tiles. This arrangement keeps the heat flow path from being short-circuited.

The tiles are cemented into position on each filler strip in order to facilitate assembly. In an assembled core; they are held in place by the filler strips which are loaded radially inward by the lateral support springs. Ledges on the filler strips limit axial movement of the tile. This arrangement serves to retain the tile in position during operation should the cement bond be lost.

Description. While the 11 configurations of the 3744 insulating tiles vary in size, they are all approximately 1/2 inch wide, 3 inches long and 1/8 inch thick. The longitudinal edges are beveled, and a 0.045 inch chamfer exists at one face of the remaining two edges. The irregular peripheral surface of the reactor core is covered by a thermal insulating layer of tiles.

Material Selection. Pyrolytic graphite is used for the insulating tile material because of its low thermal conductivity across the basal plane and its ability to operate effectively at extremely high temperatures.

Design Analysis. Maximum thermal stresses are calculated to be under 5000 psi, compared to a design strength which is greater than 8000 psi under operating conditions.

Environment. Basically, the tiles operate under compressive loading, with core side temperatures ranging from about 1050°R to 4450°R; maximum thermal gradients are in the order of 2000°.

E. FILLER BLOCKS

Design Philosophy. The filler blocks provide a circular surface on which the core band can bear and a seating surface for the dome end seal (Figure 7-10). They are keyed in place to prevent axial movement relative to the peripheral elements and are held in position radially by the core band, which is recessed into a peripheral groove. In addition to retention of the band, the bond groove keeps individual blocks from shifting if a key cement bond should fail. The blocks transfer the core band radial pressure into the core element bundle to prevent formation of inter-element gaps.

Description. The filler blocks are 4-1/2 inches long with an irregular cross section which is approximately 3/4 inch wide, they are used to transform the hexagonal element bundle to a circular core periphery. The cross section of the block closely simulates that of the corresponding filler strip assembly, except that a relief, 1/32 inch deep and just under 3-1/2 inches long, is machined into the peripheral contour. Each filler block contains a slot which is 0.135 inch deep, 0.286 inch wide and 2-1/4 inches long. This slot is centered on a face opposite the peripheral contour. There are 11 different filler block configurations prior to machining the peripheral contour, and a total of 162 blocks per reactor.

Material Selection. The filler blocks are machined from extruded Graphite G, the same high quality material which is used for the filler strips.

Environment. Under the core band, the blocks are subjected to about 5 to 10 psi at assembly and 6 to 20 psi during reactor operation. The operating temperature is approximately 1000° R.

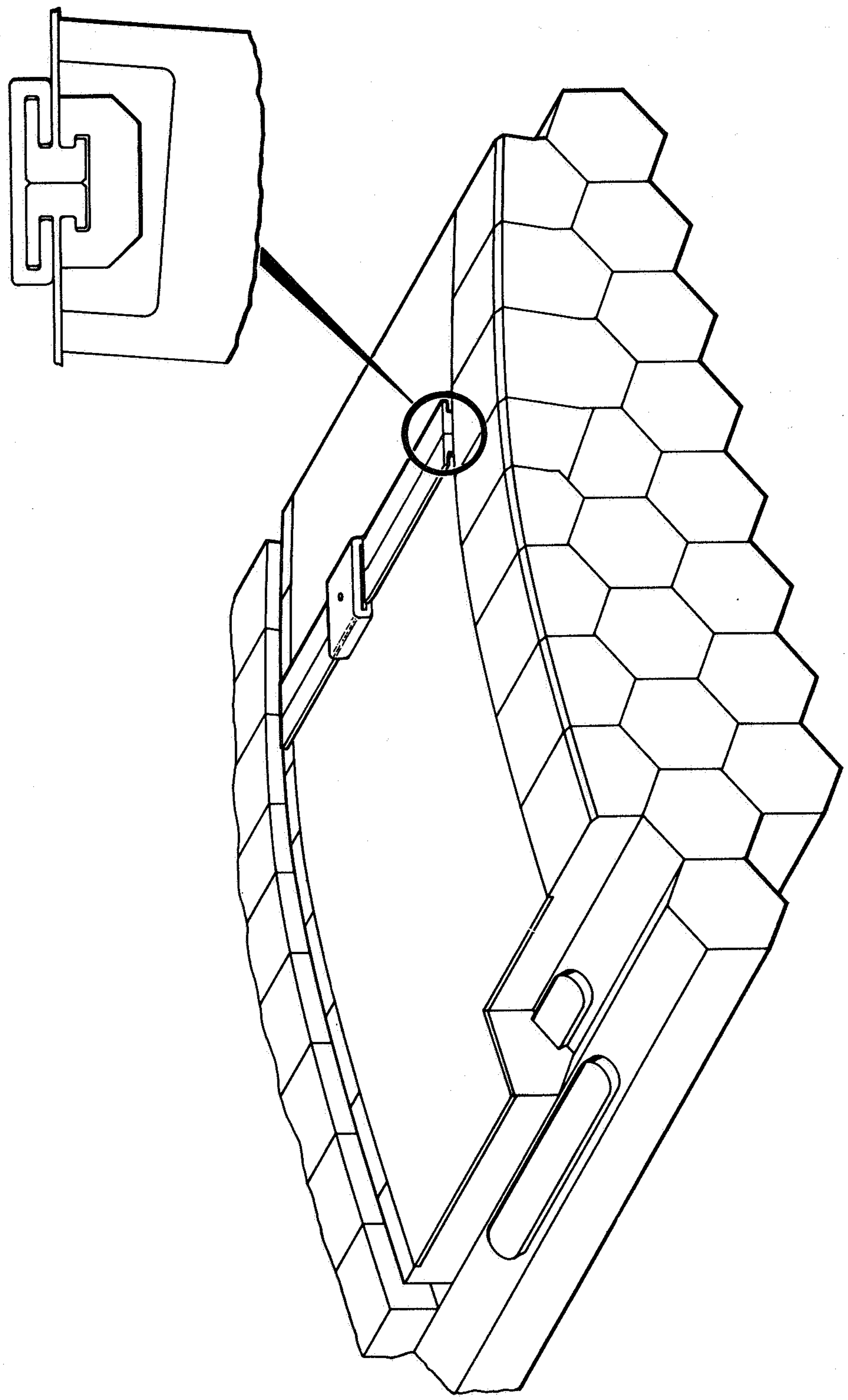

FIGURE 7-10. FILLER BLOCKS

F. FUEL CLUSTER NUT

Design Philosophy. The fuel cluster nuts fasten the fuel cluster assemblies to the core support plate and transfer the axial pressure loading of the core into the plate.

Four different fuel cluster nuts are required. Three nuts are used, in conjunction with the support plate which has a tapered thickness, to retain a constant length for the fuel cluster assembly. A fourth nut, which has a socket-head configuration, is introduced for the peripheral clusters where two nuts are too close together to allow external wrenching.

Description. The nuts, which are basically tubular in shape, are about 0.5 inch in diameter and between 2 to 4 inches in overall length (Figure 7-11). Three of these nuts have 0.625 inch hex wrenching flats (approximately 0.5 inch in length) at one end and a 0.300-inch length of 7/16-28 threads at the other end. The fourth nut has an internal 0.5 inch hexagonal socket for wrenching. All nuts have six scallops, near the wrenching end, which provide for locking device crimping.

Material Selection. Type 303 Se stainless steel was chosen because of good machinability and resistance to galling.

Design Analysis. The maximum bearing stress exerted on the fuel cluster nut is calculated as 60,000 psi. This stress is produced by the combined force of the preload spring, pressure drop and temperature differential. Since the material yield strength for this item is 100,000 psi, the nut is adequate for this application.

Environment. The operating temperature of the fuel cluster nut is between 300° and 600° R. The assembly torque provides the most significant loading of this part.

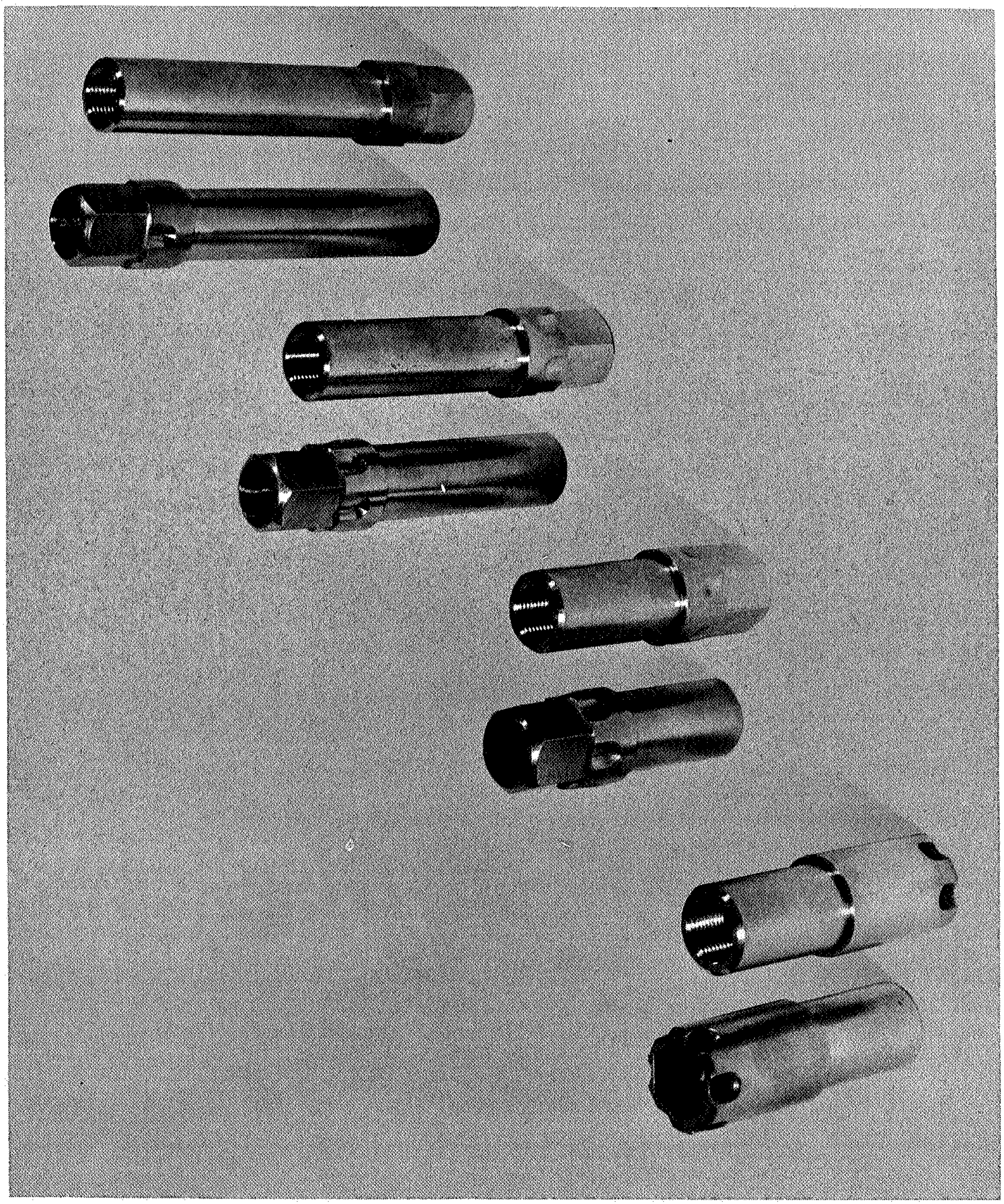

FIGURE 7-11. FUEL CLUSTER NUTS

G. CLUSTER NUT LOCKING DEVICES

<u>Design Philosophy.</u> The retention leg of the cluster nut locking device is trapped in a mating slot by inserting the device eccentric to the bolt hole centerline and then moving it radially until it is concentric (Figure 7-12). The inserted cluster nut shank maintains the device in this concentric, trapped position. The locking device is then deformed at two points into grooves in the cluster nut.

The nut locking device prevents the fuel cluster nut from "backing-out" due to vibration. It also retains the nut should fracture occur in the body.

Since the tie rods in the core periphery have adjacent element hex locations, it is necessary to incorporate special internal-wrenching nuts and dual locking devices to enclose two of these special nuts at the core support plate. The dual cup does not include a retention leg such as the one used for the single locking devices. However, since both nuts in one locking device must fail before the leg becomes necessary, the locking device still meets the retention requirements.

<u>Description.</u> The nut locking device is a 3/4-inch diameter, 1/2-inch deep, cup-shaped part with a 3/8-inch long retention leg that extends axially from a circumferential point on the closed end of the cup.

<u>Material Selection.</u> Type 303 stainless steel is used for the nut locking device because of its good machinability and cryogenic ductility. The cup is annealed so that it can be deformed into the slots of the nut.

<u>Design Analysis.</u> Experimental tests of the locking strength of this device have indicated that there is a torque capacity of 275 in.-lb. to prevent the cluster nut from backing out.

<u>Environment.</u> The cluster nut locking device is expected to reach 1000° R during pulse cooling, but during reactor operation its temperature is only around 300° R.

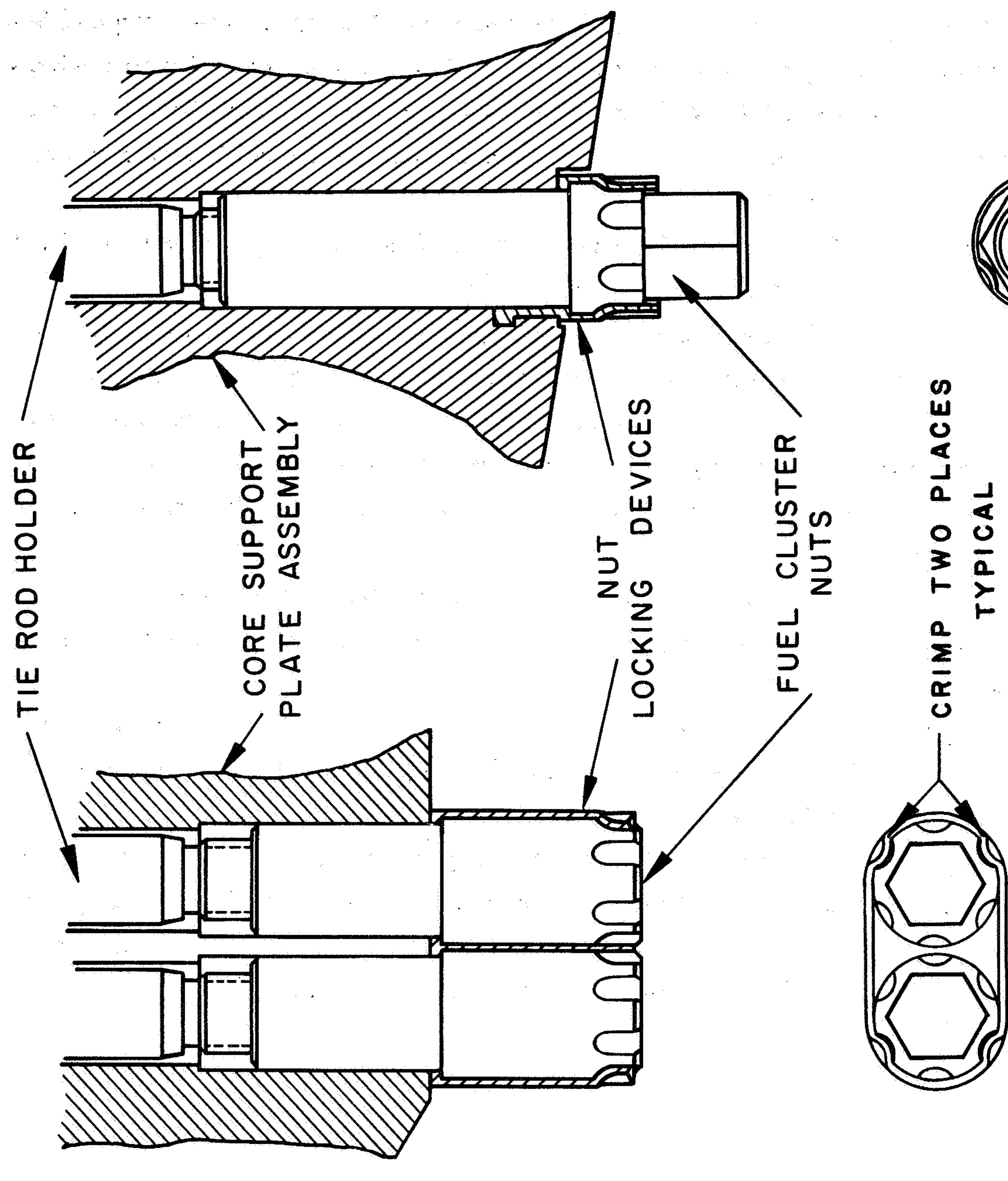

FIGURE 7-12. CLUSTER NUT LOCKING DEVICE

H. BAND

Design Philosophy. The core band is designed to provide a constant positive bundling load on the core inlet (Figure 7-13). This means that the axial leakage gaps between elements at the core inlet can be held to a minimum during assembly and will remain in that condition throughout handling and operation. The band is also used to make certain that the leakage gaps between elements at the inlet will always be less than or equal to the gaps in the remainder of the core, thereby increasing system stability against flow induced vibrations.

Description. Three 120-degree sections are coupled together to form the band. Each section is 0.020 inch thick and 3.350 inches wide, with flanges on each end to accept a C-type coupling. Each band coupling is 3.350 inches long and has a C-shaped cross section which allows it to slide over the flanges on the core side of the band segments. There are three different coupling widths, varying in increments of 0.030 inch. The variation provides a total of nine band length combinations which are needed to accommodate the tolerance accumulations of the band components and the actual core circumference.

A tab extends from the central section of the two larger couplings to help prevent rotation of the band flanges under the hoop load. A hole through the coupling cross section accepts a locking pin which retains the couplings axially on the flanges. A C-shaped locking clip traps the locking pin by sliding over the flanges on the outer diameter of the side of the band. The clip is held in place by local deformation of the flanges.

Material Selection. Both the core band and the core band couplings are made of 5A1-2.5 sn titanium alloy which has extra low interstitial content. The minimum yield strength is 90,000 psi which increases to 168,000 psi at 140° R. Elongation at room temperature is 8% which increases, as the temperature decreases, to approximately 12% at 140° R.

Design Analysis. The maximum initial pressure which the band exerts on the core is 25 psi during cold flow testing and 6 to 15 psi at full-power, steady-state operating temperature. This represents a maximum operating hoop stress in the band of 22,300 psi, which is well below the material yield stress. A detailed stress analysis of the band and coupling is presented in TME-599.

The band is cooled by flow passing between it and the adjacent lateral support segments. To insure a 6 psi preload, the core circumference must be measured under an identical preload during assembly.

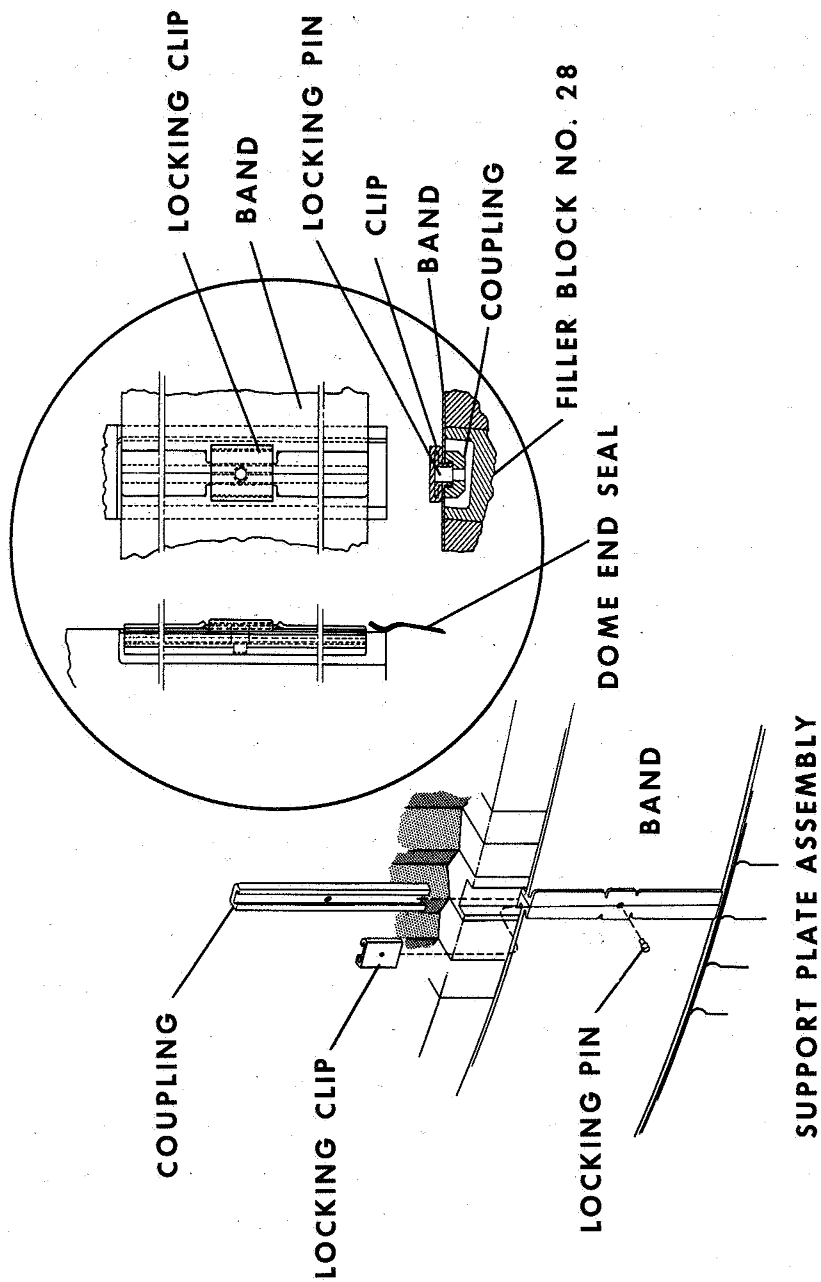

FIGURE 7-13. CORE BANDING DETAILS

Environment. During engine start-up, the band is cooled to 350°R. This temperature produces a core bundling pressure of 16.5 psi. Analysis of the operating temperature is continuing, and the band and/or filler blocks will be modified to limit the bundling pressure to a minimum of 6 psi should such modifications be necessary. This 6 psi pressure would correspond to a maximum band temperature of 600°R.

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I. CORE ASSEMBLY AXIAL SUPPORT SYSTEM

The primary function of the core assembly axial support system is to provide a mechanical support connection between the core support plate and the dome end support ring on the outer reflector. This connection allows the axial loads to be transmitted from the dome support plate to the outer reflector. These loads are primarily those of the core support plate and the core pressure drop. A secondary function is to provide a means of transmitting the inner reflector area axial load of approximately 6000 pounds at assembly and 16,800 pounds during operation. These axial loads are transmitted through the preload springs to the core support ring. The components used in this system are the core support ring assembly, barrel retaining screws, instrumentation ring, bolts and locking devices, spring support ring, preload springs and retainers, and the spring bearing ring (Figure 7-14).

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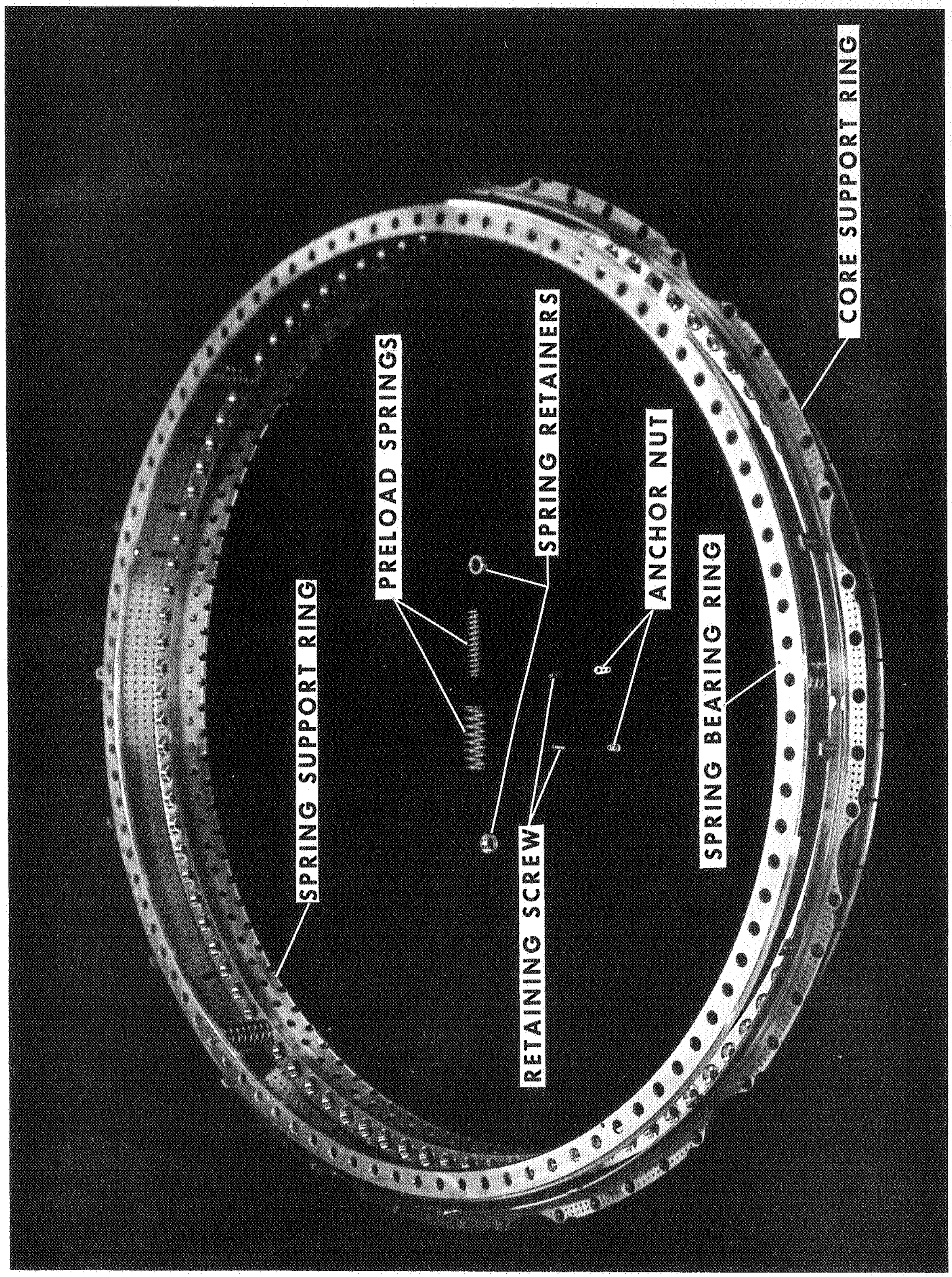

FIGURE 7-14. CORE ASSEMBLY AXIAL SUPPORT SYSTEM

1. Barrel Retaining Screw

Design Philosophy. Barrel retaining screws are used to attach the aluminum barrel to the core support ring. They carry the assembly preload force of the axial springs at the graphite cylinder until the nozzle and pressure vessel are installed. At that time, they permit the total preload force to act upon the nozzle and act only as a restraint against rotation of the aluminum barrel which, in turn, prevents rotation of the graphite cylinder.

Description. The barrel retaining screws are Philips-head shoulder bolts having an 0.190-inch diameter NF 3A thread and a 0.250-inch shank diameter. They are silver plated to prevent thread galling.

The screws fit through a 0.280-inch slot in the barrel and the 0.250-diameter shank bottoms against the core support ring. The 0.190-inch diameter shank passes through a 0.203-inch diameter hole and screws into an anchor nut riveted to the inside of the core support ring.

The barrel is slotted axially for a distance of 0.700 inch to permit relative motion of 0.45 inch between the barrel and the core support ring.

Material Selection. The barrel retaining screws are made of corrosion resistant A-286 alloy steel, per AMS 5735E. This material was selected on the basis of high strength.

Design Analysis. Assuming a conservative assembly load of 7000 pounds distributed equally over the 18 barrel retaining screws, the average shear stress has been calculated as 13,700 psi which corresponds to a peak shear stress of 18,200 psi. The allowable peak shear stress is 48,000 psi, based on the room temperature stress of 85,000 psi, the screws are acceptable.

Environment. The temperatures which this unit encounters are essentially the same as those experienced by the core support ring tabs produced by the flowing H_2.

2. Support Ring

Design Philosophy. The core support ring assembly provides the load carrying member which transmits the axial loads and some lateral loads from the core subassembly to the dome end support ring (Figure 7-15). A slotted conical shape is used to provide a flexible design which connects the two different-sized diameters of the core support plate and dome end support ring.

Bolts were chosen rather than a quick disconnect clamp for the core support ring-core support plate flange connection in order to provide a joint which can be analyzed easily and more reliably. This modification was made in conjunction with a change in the remote disassembly procedure which eliminated the process of pulling the core through the support ring.

Description. The ring is a 0.125-inch thick conical shell with flanges for bolting at each end. A 25-degree core angle provides the transition from the core support plate flange to the larger diameter dome end support ring of the outer reflector. The shell is perforated with more than 3000 coolant holes (three different sizes) and has 48 equally-spaced axial slots approximately 2.5 inches long extending from the dome end. These coolant holes provide uniform low temperatures in the ring despite the low thermal conductivity of the material. Slotting the shell provides additional flexibility to accommodate the relative thermal expansions between the ring and both the core support plate and dome end support ring without producing excessive stresses.

The dome end flange is tapped for forty-eight 1/4-inch diameter titanium bolts which join the ring to the core support plate. To eliminate galling, the bolts are plated with a layer of nickel from 0.0001 to 0.0002 inch thick, followed with a layer of silver 0.0002 to 0.0004 inch thick. An axial hole in the center of each bolt provides a channel which is used to cool the bolts. Internal wrenching heads have scalloped sections on their periphery to allow crimping of the locking device which retains the bolt.

Sixty through holes in the nozzle end flange accommodate 5/16-inch diameter titanium bolts and locking devices which secure the flange to the dome end support ring, and twelve scallops in the nozzle end flange periphery allow removal of the outer reflector control drums without parting the flange.

Extending axially from the nozzle end flange are a locating pilot and 18 tabs to which are riveted anchor nuts that carry the load of the aluminum barrel during assembly handling.

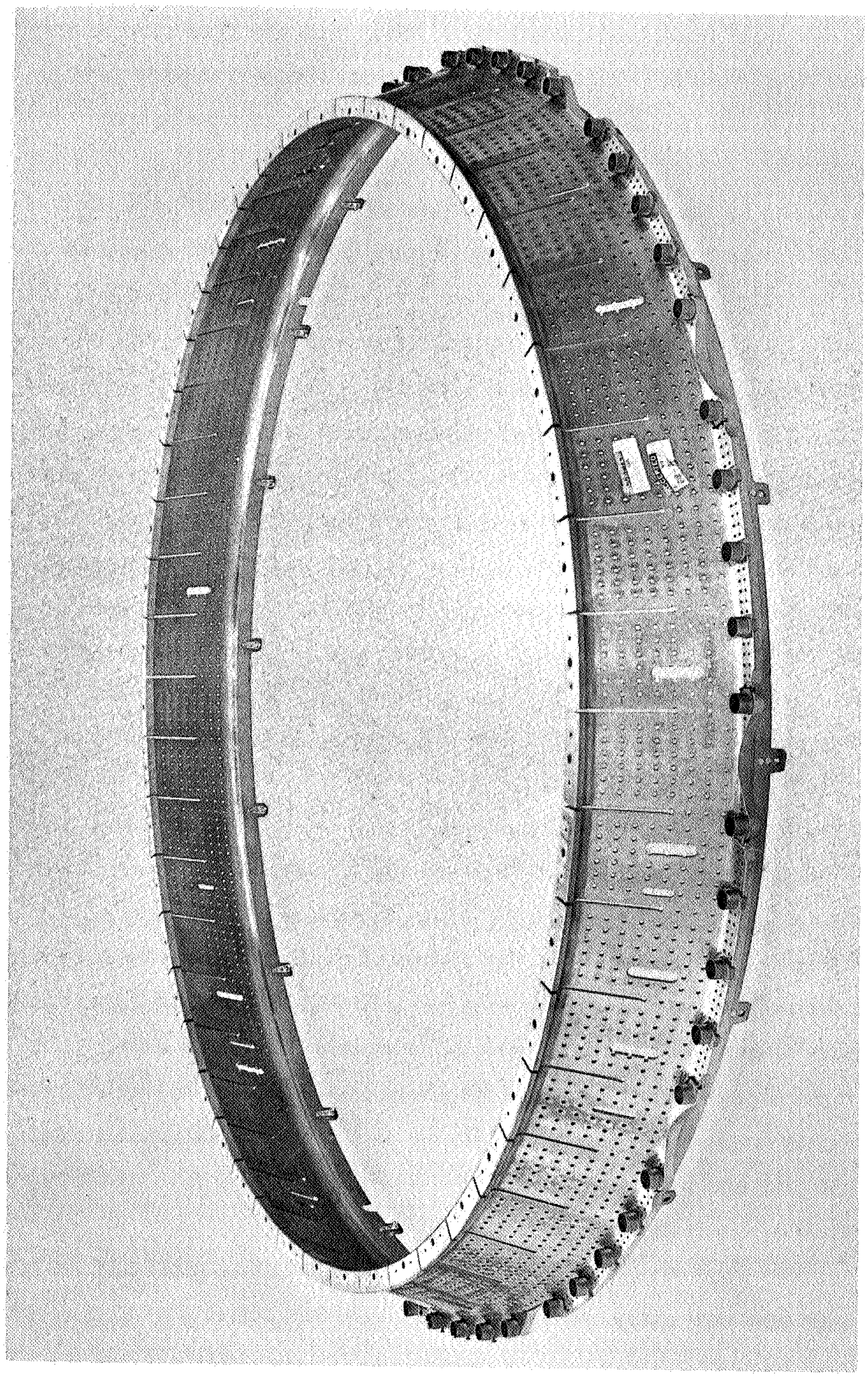

FIGURE 7-15. SUPPORT RING

Material Selection. The material selected for the core is 5Al-2.5Sn titanium alloy with extra low interstitial content. The material was chosen over others because of its high strength, low density, low elastic modulus, good ductility and low thermal coefficient of expansion. Several materials such as Inconel X, steel, and aluminum were examined to choose the most effective and most reliable considering the interaction of the materials of the aluminum core support plate, the core support ring, and the done end support ring under all phases of environmental conditions. In order to insure compatibility, the material of the dome end support and core support rings were chosen at the same time.

The bolts are also made of 5Al-2.5Sn titanium with extra low interstitial content to provide a high elongation and accommodate relative thermal expansion of mating flanges while retaining adequate loads. Maximum bolt stresses are calculated to be 55,000 psi in the 1/4-inch diameter bolts and 14,000 psi in the 5/16-inch diameter bolts.

Design Analysis. In relation to the possibility that the slotted section of the ring might buckle because of axial and lateral loads which are imposed on this component, it can be said that calculations indicate a margin of safety of 4 to 1. The maximum stress has been calculated as 41,000 psi, which is well within the allowable limits of 95,000 psi, the minimum 0.2% yield strength for the material at room temperature, and the better than 116,000 psi minimum yield strength under operating conditions.

Environment. The core support ring must carry an axial load equal to 1000 lb./in. of dome end flange, while operating in flowing hydrogen gas. This load is caused by the core pressure drop. The steady-state temperature distribution of the ring is shown in Figure 7-16. Additional loads are imposed on the ring because the core support plate shrinks approximately 0.060 inch under steady-state conditions.

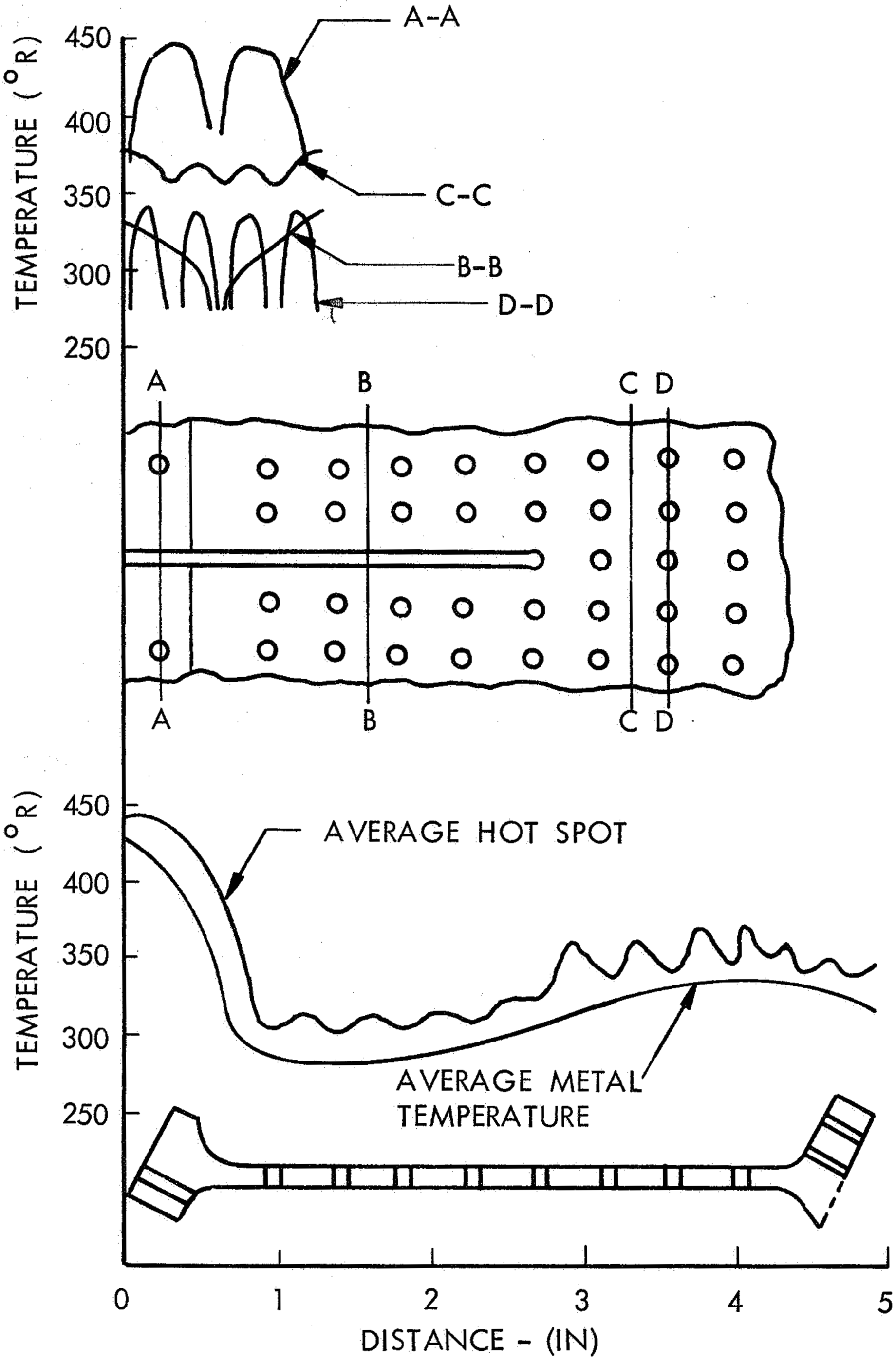

FIGURE 7-16. CORE SUPPORT RING TEMPERATURE PROFILE

3. Instrumentation Ring

Design Philosophy. The instrumentation ring serves to anchor the core instrument leads.

Description. The ring fits against the foreward face of the core support plate near the periphery. It contains radial slots on its under surface through which the instrument leads pass. The leads are trapped between the ring and the core support plate (Figure 7-17).

The ring is attached to the reactor by the core support bolts, which pass through the ring and the core support plate to thread into the core support ring. The bolt locking devices are anchored to this ring. Coolant holes at 96 places in the ring match the coolant holes of the core support plate flange. This assembly also contains the locating pins for the core inlet flow screen.

Material Selection. 6061-T6 aluminum was selected for this part on the basis of matching the coefficient of thermal expansion with the core support plate to which it is attached. In addition, aluminum has a low neutron absorption rate, internal heat generation, and less cooling than the other materials evaluated.

Design Analysis. The only loads exerted on the instrumentation ring are the bearing loads created by the bolts; these loads produce a minor compressive stress.

Environment. The ring operates in a flowing hydrogen environment of 239° R at 710 psia.

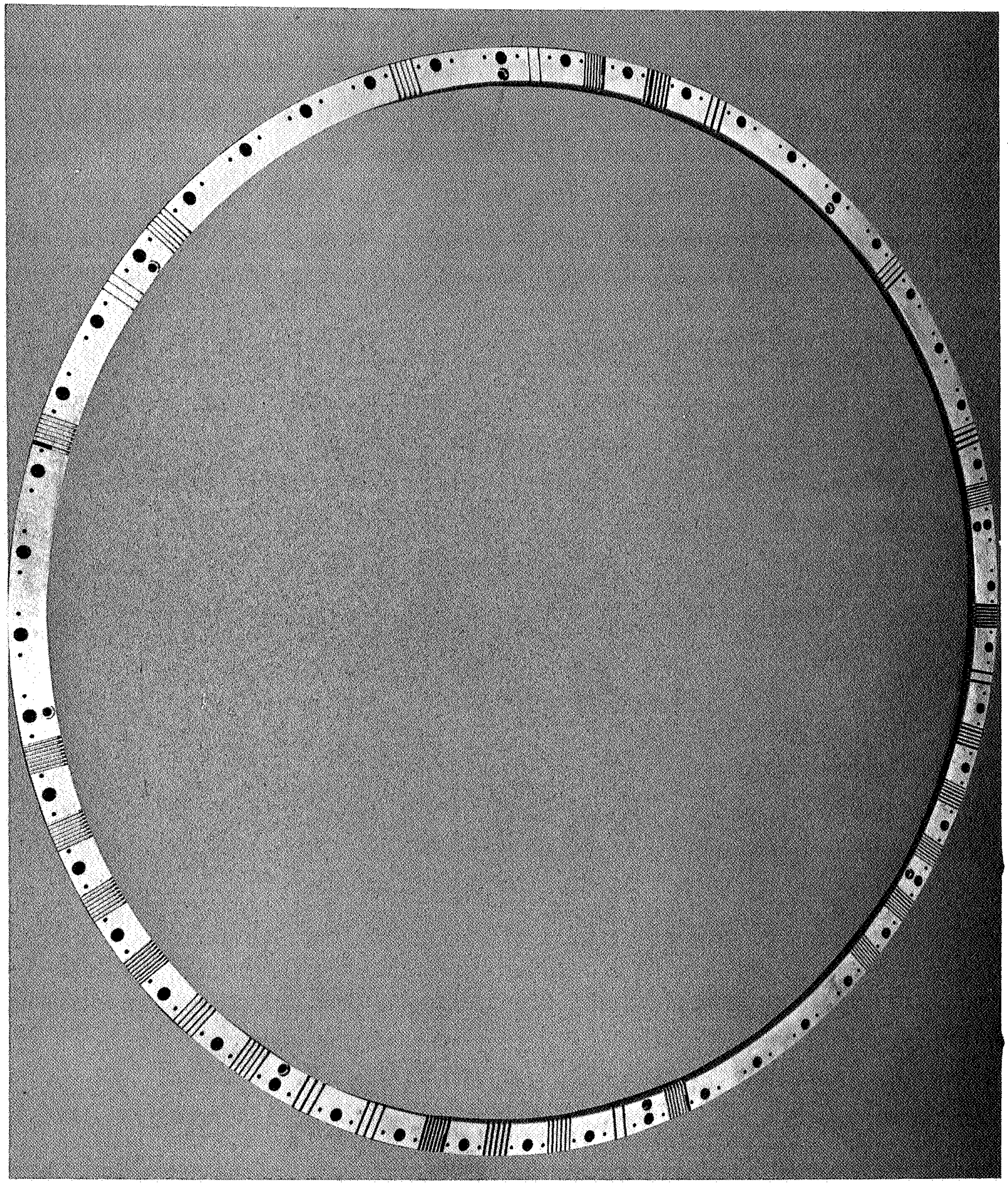

FIGURE 7-17. INSTRUMENTATION RING

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4. Locking Device

Design Philosophy. The two locking devices keep the core support bolts from rotating due to vibration and retain the bolt head in the event of bolt failure.

Description. The devices consist of a deep-dished cup with tapered tangs extending below the bottom surface. The tangs of the one locking device fit into a tapered hole in the instrumentation ring, while the tangs of the locking device fit into the core support ring. The tangs are flared out against the walls of the hole. When the bolt is installed, the shank prevents the tangs from deforming inward, thus the locking device is anchored to the reactor.

The bolt heads are scalloped around the periphery and the cup of the locking device is crimped into the recess. Since the recess does not extend to the base of the head, the crimp prevents the bolt head from leaving the cup in the event the bolt breaks. The crimp also prevents rotation of the bolt relative to the locking device. Rotation of the locking device, in turn, is prevented by a tab which hooks over the O.D. of the instrumentation ring or core support ring.

Material Selection. Stainless steel, Type 302, was selected on the basis of good ductility for forming into the bolt head locking recesses and into the tapered holes. The material has previously been used successfully on other locking devices and has been proved adequate in prototype tests.

Design Analysis. Prototype locking devices were subjected to mechanical tests to determine the adequacy of the design. As a result of these tests, the tab which prevents rotation was stiffened by increasing its thickness. Several of the prototype devices had failed because the tab bent under the torque, allowing the hooked end to ride up on the mocked-up test section.

During the test program it was found that the minimum torque required to rotate the bolt within the locking device ranged from a minimum of approximately 10 in.-lb. per crimp to a maximum of 17 in.-lb. per crimp.

As a result of these tests, it was decided that two crimps per bolt would prevent the devices from loosening.

Environment. The locking device environment is essentially the same flowing hydrogen as the core support ring.

5. Spring Support Ring

Design Philosophy. The decision to use a conical core support ring to supply axial support for the core made it necessary to devise a separate support system for the graphite cylinder. The cylinder support system requires springs of some form to accommodate relative thermal expansions and seat the cylinder against the nozzle for sealing. A base is required to transmit the chosen coil spring load to the core support ring. To meet all of these requirements, a separate spring support ring was designed. This solution was preferable to using a contoured addition to the core support ring since such an addition would have appreciably increased the titanium mass which is hard to cool. In addition, this latter arrangement would have reduced the flexibility of the core support ring and required a more complex machining on the core support ring. If a flange were added to the core support plate to supply the spring support, it would be impossible to disassemble the core from the inner reflector area by pulling the core axially through the core support ring.

Description. The spring support ring has a T-shaped cross section designed to provide a supporting base for the one end of the 192 axial preload springs while being confined to the triangular annulus between the core support plate and core support ring. Counterbored recesses in the base accept the spring retainers and thus serve to position the springs. The ring has a maximum diameter of 38.620 inches and a minimum diameter of 36.260 inches at the base and is 2.115 inches high. The 192 holes which are drilled through the cross section and the 96 scallops on the extended leg provide channels for coolant flow going from the inner reflector to the core support ring. At the same time, these holes and scallops serve to cool the ring itself. The leg extends to the acute angle formed by the core support plate and core support ring where the ring is kept in proper radial alignment by a close diametrical fit with the core support plate.

Material Selection. The ring is machined from a 2219-T852 aluminum forging. Aluminum was chosen because of its low density, low heating rate and high thermal conductivity. The 2219 alloy was chosen because it retains its mechanical properties after being exposed to limited elevated temperatures during decay heat. The use of aluminum greatly reduces relative thermal expansion in the diametral fit at the core support plate, thereby providing good alignment control for one end of the preload springs.

Design Analysis. A nominal axial load of 16,800 pounds is applied by the springs during full-power, steady-state operation. The stress in the ring associated with this load is 15,000 psi, which compares favorably with the 35,000 pis stress allowable for this material.

Environment. This ring is used to support the 16,800-pound load caused by the spring preload plus relative thermal expansion, in a flowing hydrogen environment of 239° R at 710 psia. The temperature distribution for the spring support ring is shown on Figure 7-18.

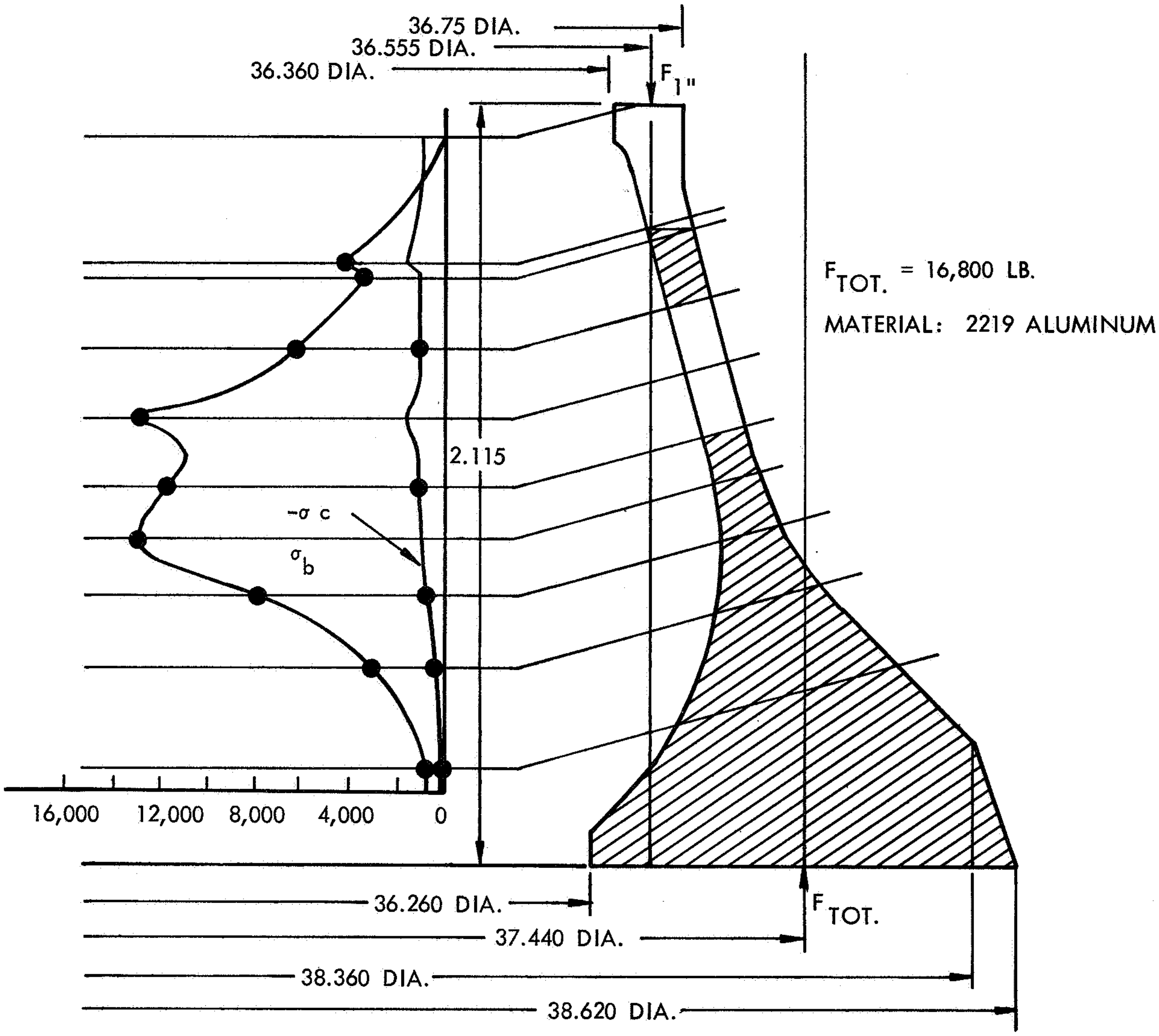

FIGURE 7-18. SPRING SUPPORT RING TEMPERATURE PROFILE

6. Preload Springs and Retainers

Design Philosophy. The function of these components is to provide axial force to the inner reflector barrel to hold it in position against the nozzle during operation, yet permit relative thermal expansion around the mechanical load bearing "loop". This loop is traced through the graphite cylinder, nozzle, pressure vessel, dome end support ring, core support ring and back to the graphite cylinder.

Other springs were tried, such as Belleville and leaf springs, but the required motion, load control and envelope could not be accommodated. Other solutions were also considered. For example, the feasibility of affixing the cylinder to the nozzle was considered, but was discarded because that arrangement would have to accommodate considerable relative thermal expansion. Again, the possibility of resting the core load directly on the graphite cylinder was also considered. That arrangement would eliminate the springs in their present location, but would have still required that they be used elsewhere in some form to accommodate negative core loads or relative thermal expansions.

Description. The preload springs are comprised of 192 coil springs. These springs are made in two diameters so that smaller spring can nest within the larger. This design produces 96 sets of coil springs which fit in the annular area between the core support ring and the inner reflector cylinder. These springs bear against the spring support and spring bearing rings. The larger diameter spring has a 1.168-inch O.D. with a 0.164-inch wire diameter producing a 124.0 lb./in. rate. The smaller diameter spring has a 0.768-inch O.D. with an 0.117-inch wire diameter producing a 71 lb./in. rate. In order to prevent coil clash, the larger spring has a left-hand coil winding, while the smaller spring has a right-hand coil winding. Two springs must be "nested" to give a sufficient load in the limited triangular space between the core support ring and core support plate.

The spring retainer is a thin-walled cylindrical cup with a 0.783-inch inner diameter and a 0.300-inch height. It extends between the larger and smaller springs to keep them properly spaced. The base of one cup recesses into a counterbore in the spring bearing ring, while a second cup recesses into a counterbore in the spring support ring, thereby providing positive radial and tangential positioning of each set of nested springs. A 0.500-inch diameter hole is cut into the base of the retainer to allow passage of coolant flow from the end of the graphite cylinder.

Material Selection. The springs are made of cold worked 302 stainless steel which was chosen because of its high allowable stress (in excess of 100,000 psi) and high elongation for a spring material at cryogenic temperatures.

The retainers are machined from 6061-T6 aluminum because of very low density and low heating rate, high thermal conductivity, ease of machining and adequate stress limits.

Design Analysis. The preload springs have a maximum stress of 90,050 psi for the larger and 91,000 psi for the smaller.

Environment. The springs and retainers operate in a flowing H_2 gas environment at 239° R and 710 psia. Maximum loads are 120 pounds for the outer spring and 70 pounds for the inner spring. The nominal load produced by the combined sets of springs is 16,800 pounds at reactor steady-state conditions.

7. Spring Bearing Ring

Deisgn Philosophy. The spring bearing ring serves as a support and locating device for the nozzle end of the 192 axial preload springs. It also transmits the axial spring load into the inner reflector graphite cylinder and serves as a metering and flow directing device as well, distributing the cooling fluid from the graphite cylinder into the spring centers.

Description. The ring, which is made of 2219 aluminum, has a cross section which is approximately 0.27 x 1.18 inch. Recesses are counter-bored into the ring at 96 locations in order to fix the position of the spring and retainer. Since the ring shrinks a certain amount relative to the graphite reflector upon which it bears, keying devices are provided to locate the ring with respect to the graphite barrel. The ring is not highly stressed in operation.

Material Selection. Aluminum was chosen for this unit because of its low density, low heating rate and high thermal conductivity. The spring bearing ring material was made the same as the spring support ring to maintain alignment of the preload springs under themal contractions.

Design Analysis. The only loads exerted on the ring are those spring bearing loads which produce a minor compressive stress.

Environment. The ring operates in a flowing hydrogen environment of 239^{o} R at 710 psia.

J. GRAPHITE CYLINDER

Design Philosophy. The graphite cylinder functions primarily as a neutron reflector between the core and the outer reflector. In addition to this reflector function, it serves as a flow barrier between core flow and reflector cooling flow, a thermal and pressure barrier between the core and the outer reflector, an absorber and transmitter for the core lateral loads and a retainer for the filler strips.

Description. The graphite cylinder is a hollow right circular cylinder approximately 54 inches in length with an I.D. of 36.1 inches and an O.D. of 39.8 inches. Various grooves, slots and holes are machined into the cylinder as shown on Figure 7-19.

There are 18 grooves machined into the inside surface of the cylinder. Each groove is approximately 1.5 inches wide and 0.6 inch deep. The aft surface of 17 of these grooves has a sealing surface with a 63 rms finish; the remaining groove houses the segmented retainer and has no sealing surface. These 18 grooves actually house both the segmented seals and the filler strip retainer.

One hundred and forty-four 0.250-inch diameter holes are drilled axially the length of the cylinder to within 0.6 inch of the interface seal surface. These are coolant flow holes which lower the temperature and thermal gradients within the graphite cylinder.

Three hundred and twenty-four contoured cutouts are provided for the lateral support spring assemblies on the outer diameter of the cylinder. These cutouts are interconnected by grooves and this arrangement furnishes a full-length axial coolant flow path which provides coolant to the spring assemblies. The cutouts are contour-machined to minimize the amount of graphite removed from the cylinder. Four holes are machined in each cutout. Two of these are tapped with 5/16-24 UNF-2B threads for the lateral support retaining screws. The remaining two holes (0.3750 diameter) are machined through the cylinder wall. These holes accept and guide the lateral support plungers. The tolerance on the diameter of the plunger holes is held to within 0.0005 inch to assure a good sliding fit while minimizing leakage between the spring assembly cutouts and the segmented seal grooves. The cylinder is made from extruded grade H4LM graphite.

Material Selection. The cylinder is made from graphite because of its high neutron reflection characteristics and its ability to withstand high temperatures.

The particular grade of graphite selected is type H4LM, manufactured by the Great Lakes Carbon Company. The physical properties of this grade are relatively low compared

FIGURE 7-19. GRAPHITE CYLINDER

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to other graphites such as ZTA and ATJ; however, the size of the material required for the inner reflector and the fact that this material was readily available dictated the choice of grades.

A continuing effort is being made to obtain a grade of graphite with higher strength and less voids. Two major graphite manufacturing companies are co-operating with WANL on this effort. The Great Lakes Carbon Company is now processing two cylinders which have been specially prepared. These cylinders are now receiving stringent quality control monitoring during every stage of the manufacture. National Carbon has proposed a program to enlarge the size of their grade RVA which is a high strength, high density molded graphite.

Design Analysis. The many functions and environments of the inner reflector cylinder, as well as its intricate machined shape, complicate its stress pattern. The pressure differential between the O.D. and I.D. tends to radially buckle the cylinder and place it in hoop compression. The preload springs place the cylinder in axial compression. During reactor operation, the aluminum barrel shrinks onto the cylinder and increases the compressive stresses in the cylinder. The temperature gradient results in compressive stresses at the inner diameter and tensile stresses at the outer diameter.

The radial pressure differential across the cylinder could be as large as 190 psi. The cylinder's first buckling mode pressure could be less than this pressure; therefore, the cylinder may deflect until it contacts the core or outer relfector. The cylinder would then become supported and would require a pressure differential of approximately 650 psi to collapse it in the next higher buckling mode. When the loads listed above are combined with the hydrostatic pressure load, all stresses are compressive, the maximum being approximately 3700 psi.

The shear strength of the graphite threads was determined by test, rather than analysis. The static load tests showed that a fine series thread is stronger than the same size coarse series thread. Since the pull out strength of the 5/16-24 UNF-28 threads is well over 500 pounds, a 320-pound screw load caused by lateral springs results in a minimum margin of safety of 2.5.

Environment. The following environmental loads were considered in designing the graphite cylinder.

(1) Core and cylinder lateral steady acceleration loads of 4 g's and axial loads of 6 g's positive and 1.2 g's negative.

(2) A pressure differential across the cylinder which varies from 0 psi at the forward end to a maximum 190 psi at the aft end.

(3) An axial preload spring load of 16,800 pounds.

(4) The aluminum barrel shrinkage pressure loads.

The cylinder's steady-state temperature distribution varies on the outer diameter from approximately 240° R at the aft end to 330° R at the forward end. Variance in the inner diameter ranges from about 1000° R at the aft end to 500° R at the forward end.

During operations, the hydrogen flow on the outer diameter of the cylinder ranges from approximately 740 psia at 120° R in the aft end to 700 psia at 200° R at the forward end. The inner diameter, in turn, ranges from 700 psia at the forward end to 550 psia at the aft end.

K. LATERAL SUPPORT SYSTEM

The lateral support system (Figure 7-20) is used to bundle the NRX-A reactor core during steady-state operation, to provide a distributed seal system about the core periphery and to absorb steady and dynamic lateral loads to and from the core during shipping and engine operation. The bundling force is evenly distributed on the core in order to minimize axial and radial coolant leakage paths between fuel elements. This sealing and load-transmitting system is composed of a series of equally-spaced rings of spring-loaded segments. Two rows of seals act to prevent the axial ejection of the filler strips. The components used to perform these functions are: seal segments, retainer segments, plungers, lateral support springs, spring brackets, and spring retainer screws.

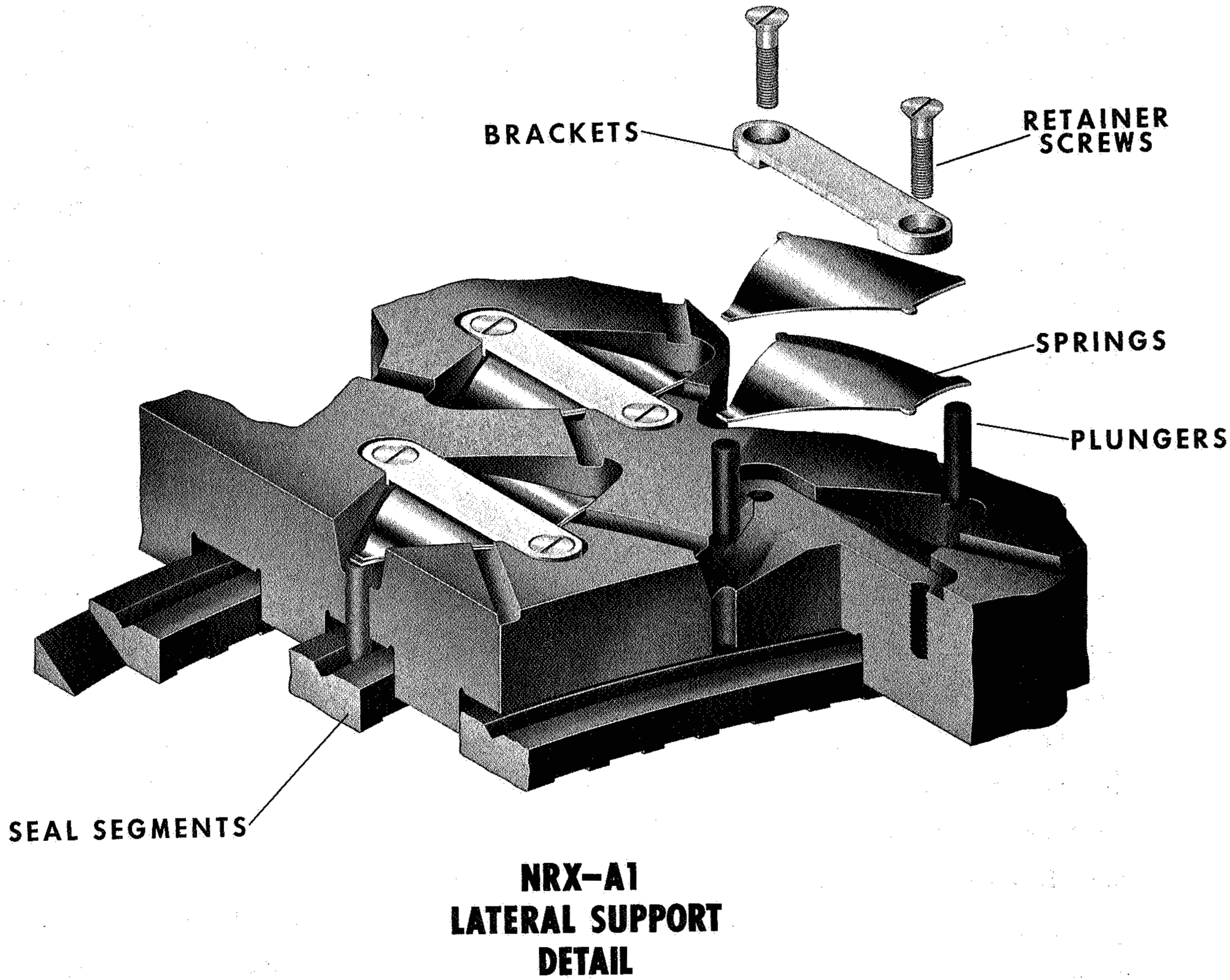


FIGURE 7-20. LATERAL SUPPORT SYSTEM

1. Seal and Retainer Segments

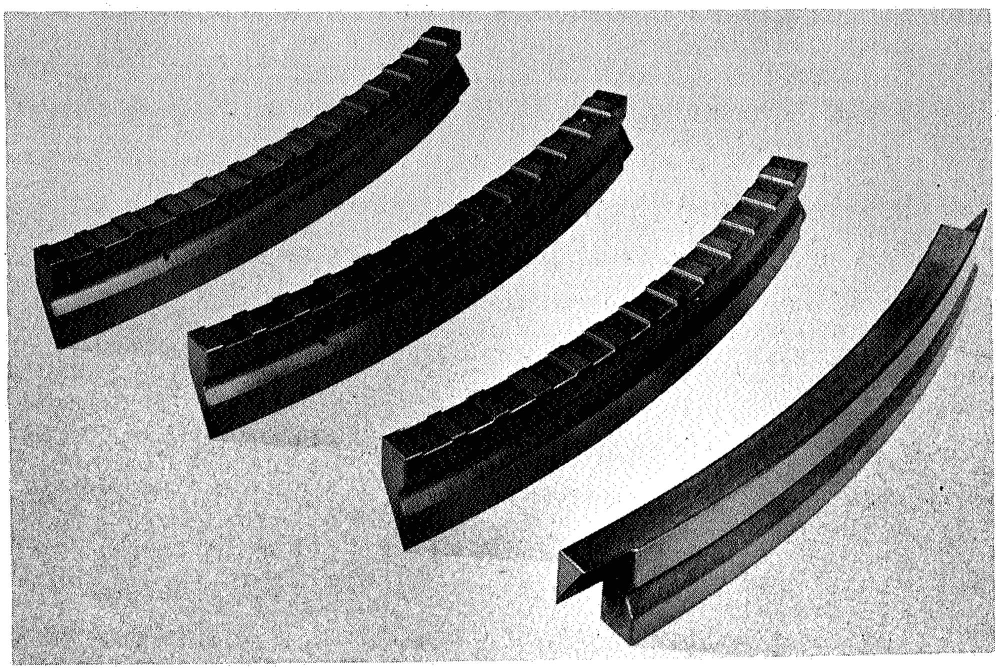
Design Philosophy. In addition to transmitting lateral loads, the seal segments prevent the high pressure component cooling flow (propellant) from by-passing the core. Sealing is accomplished in the area between the graphite cylinder and the core periphery by spring loading the segments against the core periphery and a ledge on the graphite cylinder. In this manner, the segments prevent uncontrolled axial leakage. The seal and retainer segments transmit loads from the lateral support springs and plungers to the core. Conversely, they transmit the core lateral loads to the lateral support spring system.

The segments are designed to lower the pressure of the coolant flow, in labyrinth fashion, from a pressure of approximately 700 psi at the core inlet to a pressure of approximately 550 psi at the core exit. The labyrinth-like seal was selected because of its high reliability due to the negligible effect that failure of a given seal would have on reactor operation. The segments are further designed with a set of predetermined flow grooves to control the radial bundling pressure applied to the core by controlling the pressure of gas flowing past the seals. In this manner, an optimum pressure bundling profile is obtained.

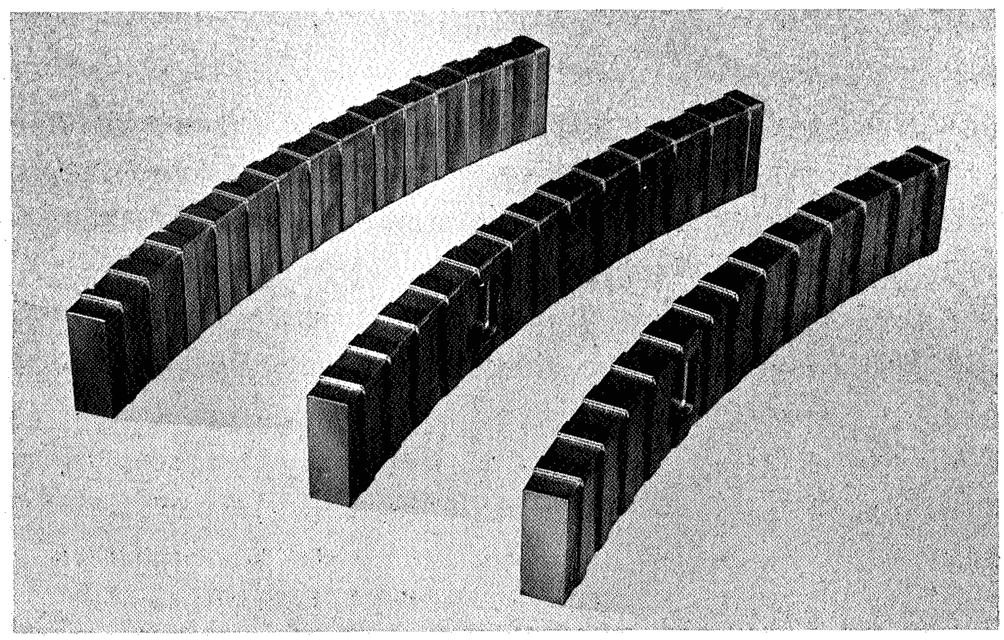
Description. In all, there are 16 active seal rows, one inactive seal row and one retainer row, each row having 12 segments. Eighteen rows were selected in accord with the accompanying component geometries (i.e., lateral support springs, graphite cylinder and spring retaining brackets) rather than on the basis of design analysis. Figure 7-21 shows the seal and retainer segments.

For the purposes of this report, the seal rows are numbered 1 through 18, with row 1 at the forward end and row 18 at the aft end. Row 1 is an inactive seal row of segments which bears against the core band. Row 17 is a row of retainer segments which locates and retains the filler strips. The remaining 16 rows are active seal rows.

Each seal segment consists of a 32° 20' segment of a circle with a radius of approximately 18 inches. Approximately 0.7 inch of the end of each segment is prepared in a manner similar to the overlapping ends of a single ring piston ring. When adjacent segments are joined, these ends overlap and result in a joint with no direct leakage path. However, a clearance of 0.006 to 0.010 inch is designed into the mating angled surfaces. This clearance prevents the seal from being lifted from the seal surfaces or breaking at the ends due to interference caused by thermal gradients, core growth and relative motion between adjacent seal segments.



SEAL SEGMENTS



RETAINER SEGMENTS

FIGURE 7-21. SEAL AND RETAINER SEGMENTS

A letter "D" is engraved on the surface of each segment to indicate which surface should face the dome end. This procedure ensures proper orientation during installation. The sealing surfaces have been provided with a 63 rms finish to ensure adequate sealing.

A circumferential "V" groove is provided in the back of each seal segment at the location of the lateral support plungers. The purpose of the groove is to positively seal the seal segments on the seal surfaces of the inner reflector cylinder at assembly. Seating is accomplished by the component force acting in the direction of the nozzle. This force is produced by the tapered end of the plunger bearing on the "V" groove in the segments. The seating force provides positive seating of the seal against the reflector for coefficients of friction up to 0.27 without requiring the assistance of a pressure drop force.

The sealing surfaces of the segments which close against the core are provided with grooves machined parallel to the axis of the reactor. The purpose of these grooves is to adjust the peripheral pressure distribution by controlling the leakage past the seals. The depths of the grooves are as follows: Row 1, 0.060 inch; Rows 2 through 6, 0.011 inch; Rows 7 through 11, 0.009 inch; Rows 12 through 16 and Row 18, 0.007 inch; Row 17, 0.050 inch.

Every other segment in each row (except for Row 1) is provided with a 0.875 x 0.700 x 0.200-inch deep rectangular hole located in the sealing face of the seals. Therefore, a rectangular hole exists every 60 degrees circumferentially. These holes fit over a pin in the filler strip (Figure 7-22) to assure proper positioning of the assembled seal ring and to prevent the ring segments from rotating during operation. This arrangement is needed in order to prevent a plunger from lining up with a seal segment joint, where it could possibly slip into the joint and jam the seal.

Material Selection. Graphite was selected as the segment material because of the temperature environments involved. Type ZTA molded graphite has been selected over ATJ graphite as the material for the seal segments. ZTA graphite is an anisotropic material and, with the basal planes oriented as they are in the seal segment, its flexual strength is approximately 38% higher than that of the ATJ graphite. In addition, the circumferential thermal distortion is about 1/4, the bending stress due to a thermal gradient approximately 1/2, and porosity about 1/360 of that of the ATJ graphite material.

Design Analysis. During reactor operation, the seals are subjected to minor radial and axial temperature gradients. These gradients tend to make the seal's radius of curvature smaller than that of the core. The opposing action of the lateral support plungers places

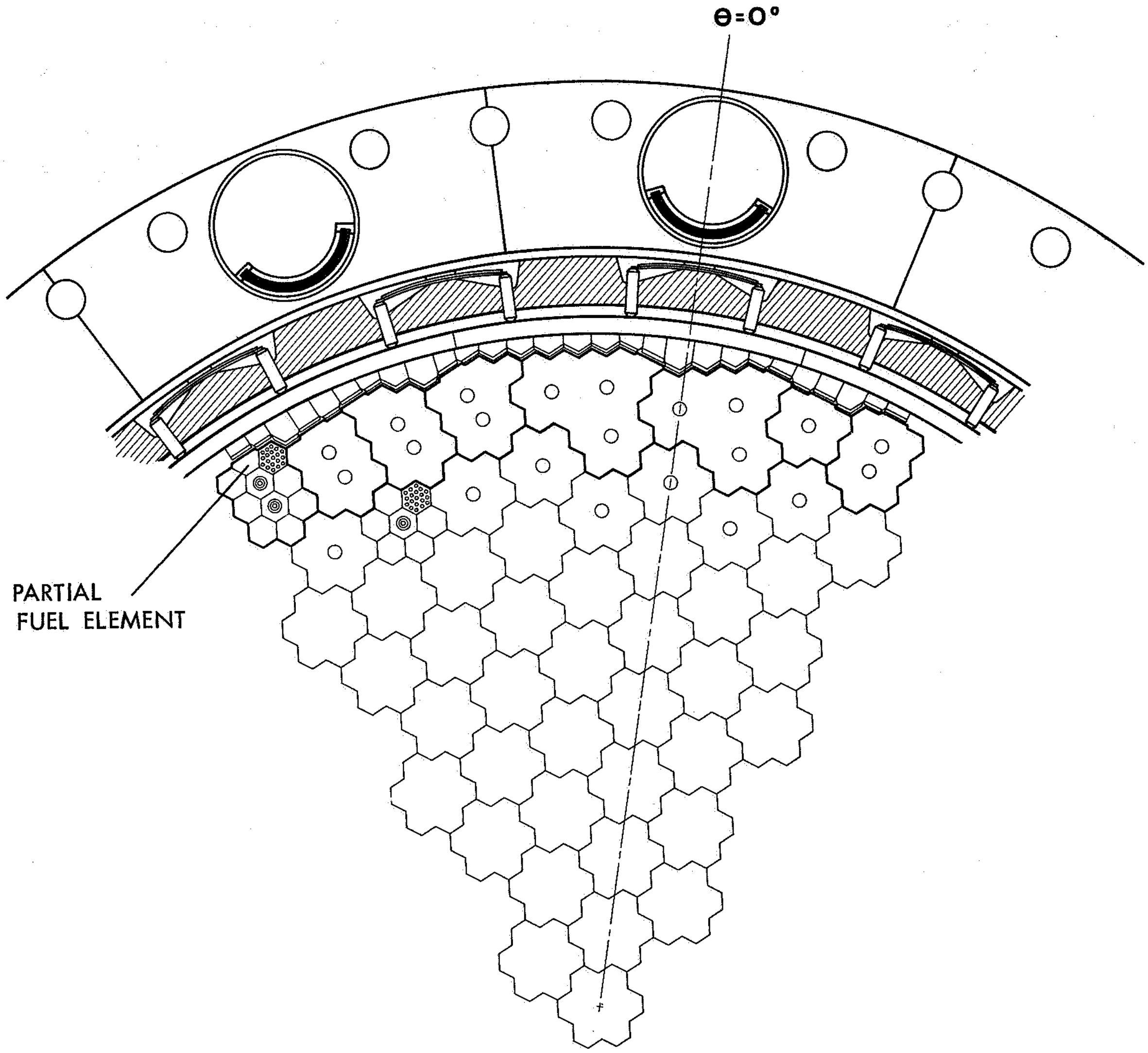


FIGURE 7-22. SECTION OF REACTOR LOOKING AFT

the O.D. of the segments in compression and the I.D. in tension. The maximum stress in the seal sector is less than 500 psi, which is well below the conservative room temperature strength of 2500 psi.

Environment. The seal segments operate in the following environment:

(1) A core lateral load of 4 g's which produces a plunger load of 62 pounds at each of the three plunger contact points on a segment.

(2) The segments are bathed by gaseous hydrogen at approximately 200 to 1200° R and 550 to 700 psia. The maximum seal temperatures vary from 500° R to 2100° R, with a radial gradient of approximately 120° R.

2. Plunger

Design Philosophy. The function of the plunger is to transmit the bundling force from the lateral support springs to the core, to transmit the core lateral loads to the lateral support springs, and to positively seat the seal segments.

The plunger is designed with the smallest possible diameter consistant with strength requirements in order to limit the flow annulus around the plunger. To further minimize the flow annulus, the dimetrical tolerance of the plunger is held within 0.0004 of an inch. Both these conditions must be met in order to minimize the volume of cold gas leaking past the plungers (from the lateral support spring chambers to the core-graphite cylinder annulus). Excessive leakage could cause severe thermal shock to the filler strips.

The finish of the cylindrical surface of the plunger is held to 16 rms to minimize sliding friction and possible jamming.

Description. There are two types of plungers; one type is used with the seal segments, the other with the retainer segment row nearest the nozzle.(Figure 7-23).

The plungers used with the retainer segments are right circular cylinders; 1.22 inches long and 0.375 inch in diameter. The ends of these items are chamfered. The plungers used with the seal segments are also right circular cylinders; however, these units, which are approximately 1.3 inches long by 0.375 inch in diameter, have one chamfered and one conoid end. The conoid ends bear against a "V" groove in the back of the seal segments. The lateral support spring thus produces a component of force in the direction of the nozzle which seats the seal segments against the nozzle end of the seal recesses in the graphite cylinder.

There are 648 plungers per reactor.

Material Selection. The plungers are machined from extruded, high density graphite. The longitudinal axis of the plunger is aligned with the axis of extrusion in order to obtain maximum bending strength and assure uniform radial growth in any direction.

The material designation is PDS 30022-3 which is the unfueled element manufactured by Westinghouse's Cheswick plant. This material was selected because it consistantly exhibited higher strength than either commercially extruded graphite rod or ATJ molded graphite during a series of three-point bend tests. Efforts are being made to obtain this same material from a commercial manufacturer.

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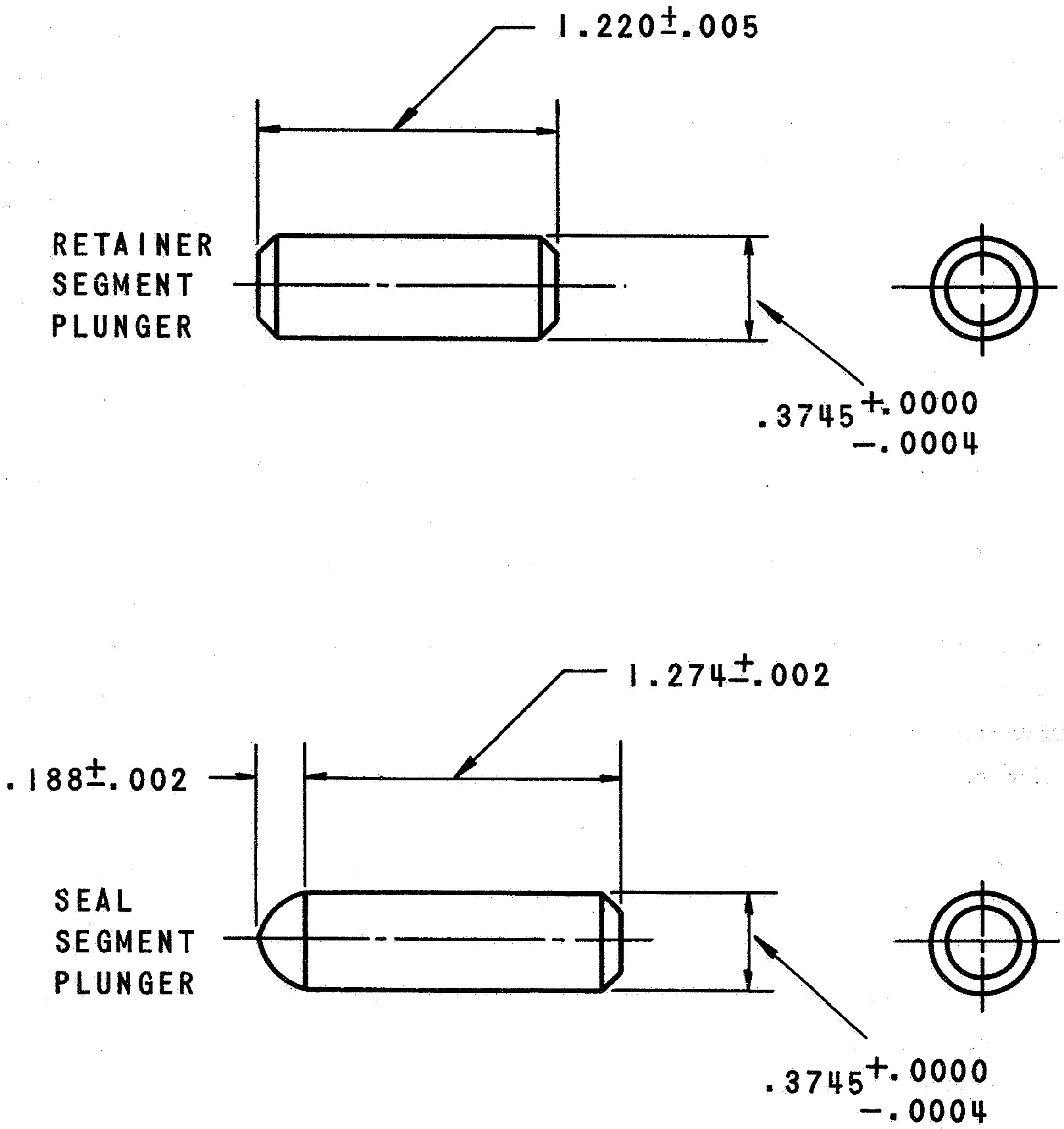

FIGURE 7-23. PLUNGERS

Design Analysis. For the worst-possible conditions, the plungers with both ends chamfered could have a maximum bending stress of 2200 psi. Since the tested value of the bending stress is well above 5000 psi, a large margin of safety exists.

The plunger with one chamfered end and one conoid end is exposed to higher bending stresses, about 4300 psi at the most extreme condition, but is again well below the breaking limits.

Environment. Core steady lateral loads of 4.0 g acceleration produce steady-bending loads of approximately 35 pounds and steady compressive loads of approximately 62 pounds. Gaseous hydrogen surrounds the plungers.

3. Springs

Design Philosophy. The lateral support springs are designed to absorb steady and dynamic lateral core loads during shipping, boost and engine operation. They are also used to bundle the core during steady-state operation.

Leaf springs have been chosen over other designs for two reasons. First, it is one of the few forms that allow the use of a small diameter plunger through the graphite. This, in turn, limits the plunger annulas area and reduces the amount of inward coolant leakage. Second, leaf springs are easily cooled because they can be placed closer to the coolant flow annulus between the graphite cylinder and outer reflector.

The spring rate is that which is required to limit the core deflection to 0.200 inch when the core is under a 4 g steady lateral load, using a 0.250-inch preload deflection on the spring. The 0.250-inch preload deflection permits maintenance of a minimum positive 7-pound spring load at all circumferential points when the core shifts 0.200 inch under the 4 g load. These values are such that the plungers and seal segments will follow the core motion at all times. The rate and load thus determined result in a 34-pound load per spring when centered at room temperature, and a peak load of 62 pounds per spring at the 0.200-inch maximum deflection. These loads are equivalent to 3.8 psi and 6.8 psi, respectively, on the core.

Description. Two leaf springs are used for each lateral support spring assembly. Each spring consists of an 0.032-inch thick stainless steel strip, cold rolled to the full-hard condition. Each spring assembly is centrally supported so that each half acts as a cantilever spring against a plunger.

The spring is diamond shaped, with diagonals of approximately 2.5 and 3.5 inches. Small tabs are provided at the ends of the 2.5 inch diagonal for positive location when assembled to the graphite cylinder. The tabs fit into grooves in the spring bracket. The leaves fit one over the other and are precurved so that, when fully loaded, they are nearly flat. Adjacent component geometry and design symmetry rather than analytical reasons, dictate the curvature of the leaves. The diamond shape yields a virtually constant-stress design which is the most economical use of metal at the core periphery. Figure 7-24 shows the springs as well as brackets and retaining screws.

Material Selection. Type 301 stainless steel is used for the springs. The 300 series stainless steel has been selected because of its resistance to ductility temperature transition

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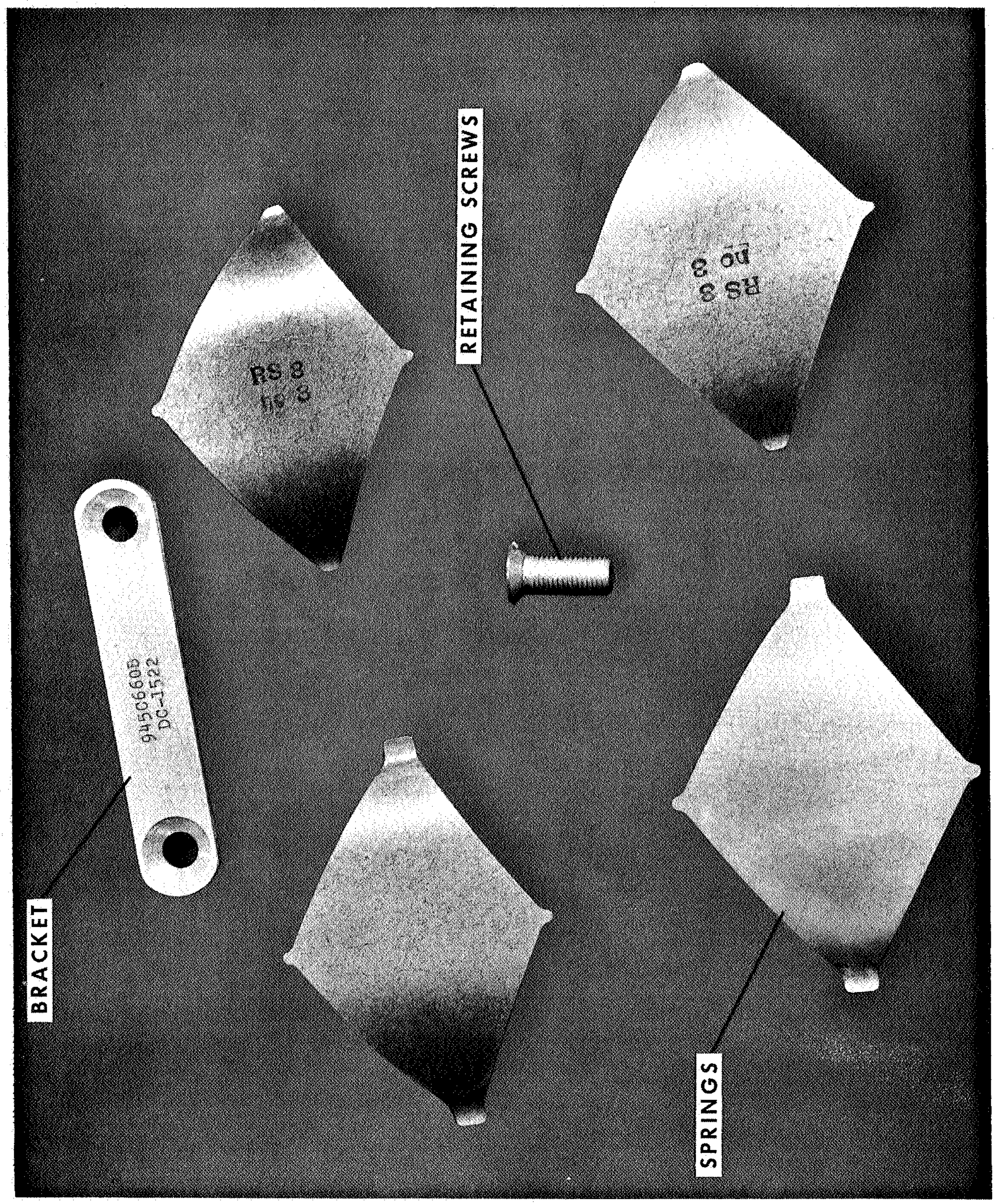

FIGURE 7-24. SPRING, BRACKET AND RETAINING SCREWS: LATERAL SUPPORT SYSTEM

in a radiation atmosphere. Type 301, full-hard, was selected over Type 302, full-hard, because of its slightly higher strength.

Design Analysis. The maximum stress in the lateral support springs is approximately 150,000 psi and the minimum yield strength for this material is approximately 180,000 psi at room temperature; this yield strength increases at lower temperatures. While the full load stress is relatively close to the yield point, it is considered better to accept this high stress and the possibility of yielding rather than introduce an additional leaf and place more steel at the core periphery.

Environment. The environment in which the spring operates produces a maximum load of 61.7 pounds per spring, either at room or operating temperatures. The spring operating temperatures range from 100° R to 200° R. Gaseous hydrogen at 100° R to 200° R and approximately 730 psi flows around the springs.

4. Spring Brackets

Design Philosophy. These brackets are used to retain the lateral support springs in such a way that holes do not have to be drilled in the springs. These holes would have caused additional stresses (Figure 7-24).

Description. The spring brackets are rectangular bars with rounded ends. They have the following approximate dimensions: length 3.875 inches, width 0.750 inch, thickness 0.166 inch. A countersunk hole is drilled at each end for the flat head fastening screws. The center portion of the bracket is recessed to fit over the lateral support spring so that approximately 0.050 inch of the width is in contact with the springs. A 0.010-inch deep by 0.400-inch wide slot is provided under each countersink to locate the load reaction points exactly and minimize the possibility of unseating the bracket with respect to the graphite cylinder.

Material Selection. The brackets are manufactured from 2219 T87 aluminum. Aluminum has been selected over steel to minimize the amount of steel at the core periphery for neutronic reasons. Type 2219 T87 has been specifically selected both because of its high strength, 57,000 psi yield strength, and its good mechanical property retention after being exposed to elevated temperatures.

Design Analysis. At assembly, the maximum bending stress induced in the bracket is approximately 25,000 psi. Since the room temperature yield point of the material is 57,000 psi, the bar will not yield. The corresponding load deflection is approximately 0.010 inch. It should also be noted that the presence of the barrel during operation will limit deflection and stresses.

Environment. The spring bracket's environment produces core lateral loads equivalent to 163.4 pounds per bracket. The operating temperatures range from 100° R to 200° R. Gaseous hydrogen at 100° R to 200° R and approximately 730 psi flows over the bracket.

5. Spring Retaining Screws

Design Philosophy. Screws are used to fasten the lateral support spring brackets and springs to the graphite cylinder (Figure 7-24). They are designed to provide the graphite threads with adequate strength in resisting lateral loads and to minimize the metal volume of the spring bracket and screw.

Description. The screws are an aluminum alloy, 5/16-inch flat head machine screw. The recess angle is 100 degrees, while the overall length is 0.875 inch. The 5/16-24 UNF threads are manufactured in accordance to MIL-S-7742.

Material Selection. The screws are manufactured from 6061-T6 aluminum which was selected over steel to minimize the amount of steel at the core periphery. Type 6061-T6 aluminum was specifically selected because its room temperature yield strength was in excess of 40,000 psi and the material was commercially available.

Design Analysis. The stress in the screw at the maximum load of 320 pounds is approximately 6000 psi, which is well below the yield point of the material.

Tests of the screw threads show that their minimum graphite proportioned limit is above 650 pounds, and their minimum ultimate load is 800 pounds.

Environment. The environment in which the screw operates produces a maximum load of 320 pounds per screw. The operating temperature varies from 100° R to 250° R and gaseous hydrogen, at temperatures ranging from 100° R to 200° R at a pressure of 730 psi, flows over the head of the screw.

L. NOZZLE INTERFACE SEAL

Design Philosophy. The nozzle interface seal is designed to provide a positive seal at the graphite cylinder-to-nozzle interface. This seal prevents coolant flow from bypassing the core and going directly from the nozzle tube bundle to the core exhaust plenum. Reliance upon a flat face contact between the graphite cylinder and the metal nozzle without a separate seal could permit excessive leakage under dimensional variations, thermal distortions, or low axial loading force.

Description. Two flexible U-shaped rings are welded together over a central stiffening ring to provide an elastic face seal between the end of the graphite cylinder and the nozzle surface. The rings are made of Inconel X-750 sheet, which is 0.022 inch thick. The insertion of the nozzle against the seating force of the graphite cylinder axial preload springs at assembly compresses the seal within the confines of a machined groove on the nozzle end of the cylinder (Figure 7-25). If a gap develops between the cylinder and nozzle, the full pressure drop from the reactor inlet to core exhaust would be exerted on the seal. This pressure acts to spread the U-shaped sections, forcing them against the sealing surfaces. The center stiffening ring prevents radial buckling of the seal. A notched flange on the ring extends between the aluminum support barrel flange and the inner reflector cylinder and retains the seal when the nozzle is removed. The notches in the flange allow pressure communication and coolant circulation to the U-ring which is seated against the cylinder.

Material Selection. Inconel X-750 was chosen as the seal material because of its high strength at elevated temperatures, its well defined fabrication techniques and the availability of data as to its properties after irradiation. However, Inconel 718 is under examination as a possible seal material which would maximize preload deflection without yielding. The greater the deflection prior to yielding, the larger the margin against possible distortions of the interface before the seal opens up. Flow tests are being performed on the seal to verify the sealing adequacy for various pressure drops and interface distortions.

Design Analysis. The seal spring has been analyzed for nominal dimensions and compressed from a free axial length of 0.396 inch to 0.300 inch. The maximum bending stress, which occurs in the U-shaped section, amounts to 85,000 psi in the compressed condition. At the same time, a force of 13.0 lb./in. of circumference, or 1500 pound total, is exerted by the seal on the inner reflector cylinder and the nozzle.

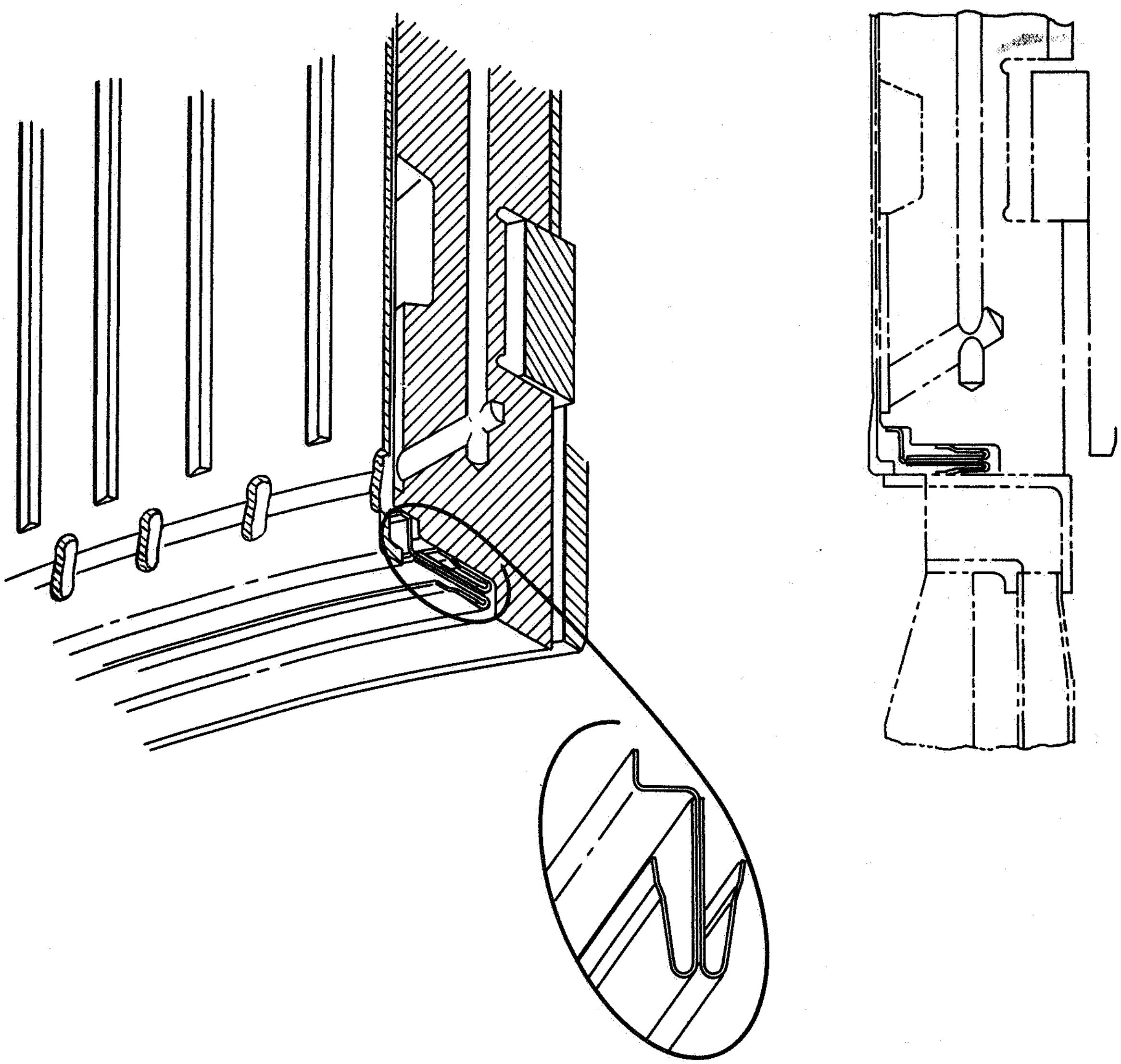

FIGURE 7-25. NOZZLE INTERFACE SEAL

It is estimated that variations in dimensions from nominal, notably variations in material thickness, could increase the stress and the load by as much as 30%. This could cause slight, but acceptable, localized yielding.

Temperature distribution in the seal has been calculated and is presented in Figure 7-26. The analysis of thermal stress and the stress caused by pressure difference has not been performed as yet.

Environment. The temperature distributions in the seal are shown in Figure 7-26. Environmental conditions at the seal produce a maximum pressure drop across the seal of 190 psi.

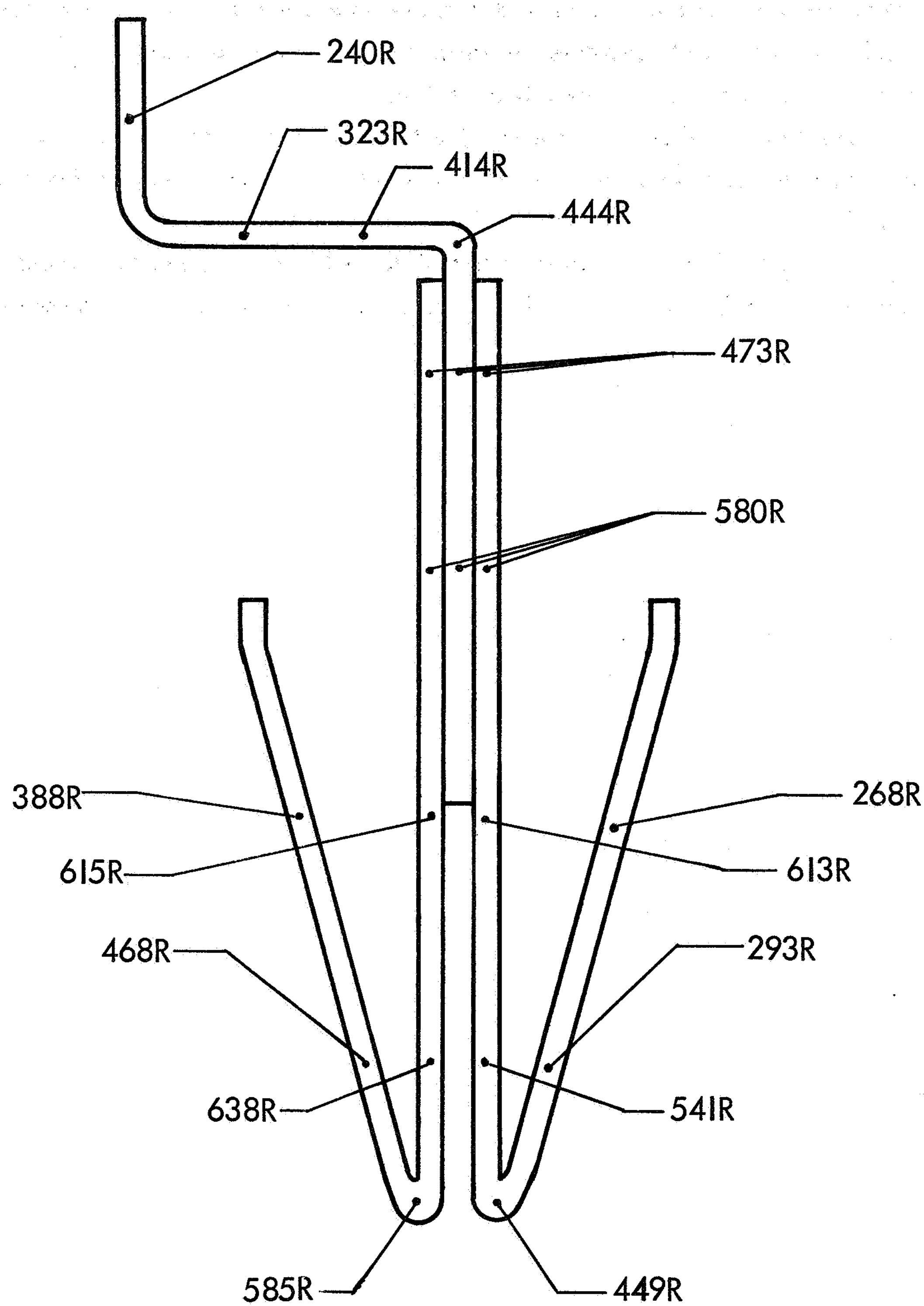

FIGURE 7-26. NOZZLE INTERFACE SEAL TEMPERATURE DISTRIBUTION

M. ALUMINUM BARREL

Design Philosophy. The barrel design was chosen to provide a handling device for the inner reflector area components which would retain the inner reflector area components when the core assembly is handled as a unit. The choice of a cylindrical design provides several additional desirable characteristics: it can be used to provide a protective sheath over the easily damaged graphite cylinder; it can be used to induce compressive stresses into the cylinder to help offset undesirable tensile stresses due to thermal gradients; it can provide a convenient flow divider for cooling the lateral support springs; it can serve to contain lateral support components in case of failure, while still allowing them to function at some reduced degree; and it can readily serve as a controlling spacer and lateral load transmitter from the inner reflector to the outer reflector.

Description. The aluminum barrel (Figure 7-27) is a thin-walled cylinder with an L-shaped flange on the aft end and 18 tabs on the forward end. The barrel is constructed by rolling 0.063-inch sheet into a cylinder and welding the ends. A circumferential weld attaches the thicker aft end flange to the base cylinder. Thirty-six axial spacers are spot welded to the outer diameter of the cylinder to insure a minimum flow gap between the barrel and outer reflector for coolant flow distribution. The spacers also provide the limiting stop against lateral motion of the graphite cylinder. Elongated holes in each tab at the forward end accommodate the radial shoulder screws which transmit the axial assembly loads of the inner reflector area graphite and the partially compressed preload springs to the core support ring. Elongating the holes allows the barrel to axially displace when the nozzle displaces the inner reflector cylinder in further loading the preload springs. The aft end flange of the barrel has a lip on the inner diameter which provides axial support for the inner reflector area components when the nozzle is not in place. The flange also serves to trap the nozzle interface seal.

A key riveted to the inner diameter of the barrel at the aft end engages a groove in the graphite cylinder and thus provides a restraint against relative tangential motion of the cylinder with respect to the core and barrel.

A metering ring is circumferentially spot welded to the forward end of the barrel to provide a regulating impedance for flow between the barrel and outer reflector.

Material Selection. The material chosen for the barrel is 2219 aluminum, which is welded in the T-31 condition and then re-solutioned and aged to the T-62 condition.

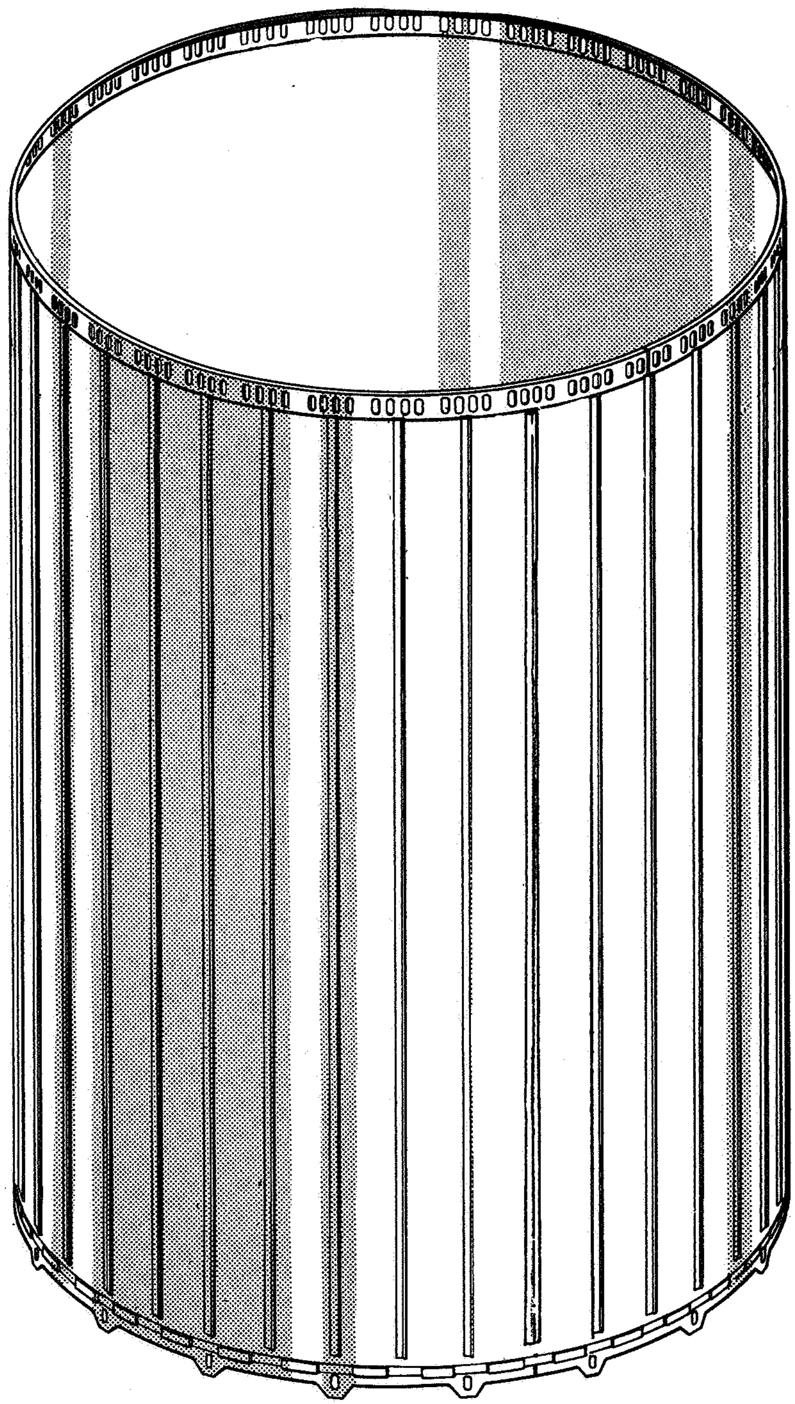

FIGURE 7-27. ALUMINUM BARREL

Aluminum provides an easily cooled, low neutron absorbing material which can be introduced acceptably between the core and control drums. Steels and Inconel X were discarded because their higher neutron absorption reduced the effectiveness of the control drum.

This specific grade of aluminum was chosen because of its high strength, "weldability" and good recovery from elevated temperatures during decay heat when compared to the other aluminum alloys. The re-solutioning heat treatment is used to produce high strength and elongation characteristics in the welds which are equivalent to those produced in the base material. At room temperatures, the base material has a room temperature yield strength in excess of 40,000 psi and a room temperature elongation of 10%.

Design Analysis. The stresses in the barrel are caused by the thermal shrinkage and pressure drop across the barrel-cylinder combination. The reactor hot operating conditions shows this stress to be approximately 16,000 psi hoop. In a cold flow test; however, this stress could be approximately 19,000 psi in both hoop and axial tension, based on a 0.3 coefficient of friction. Superimposed on the shrinkage stress, is the stress caused by a pressure drop across the barrel in the vicinity of the lateral support spring cutouts. For the present design flow routing, this added stress is not expected to be more than 3000 psi local tension. Based on the yield strength specifications covering this material, these stresses are acceptable.

Environment. The barrel operates in a flowing hydrogen environment which produces a temperature of 190° R at the aft end and 255° R at the forward end under hot operating conditions. A peak temperature of 285° R occurs during operation at approximately 17 inches from the forward end. Under cold flow tests, it is estimated that barrel temperature could go down to a minimum 50° R. The hot test hydrogen pressure distribution is 750 psia at the aft end and 710 psia at the forward end. The drop is assumed linear through the spring chambers on the barrel inner diameter.

The barrel will have an axial load of approximately 6000 pounds during handling. This load is a combination of spring preload, graphite cylinder and lateral support system weights.

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